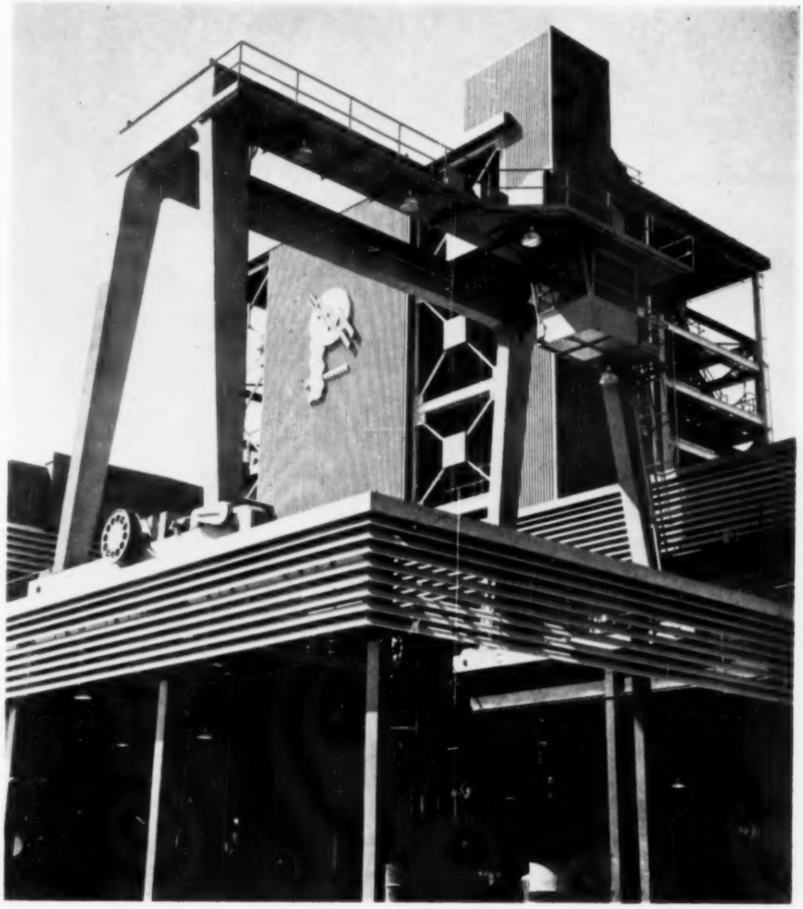
Combustion

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION



May 1960

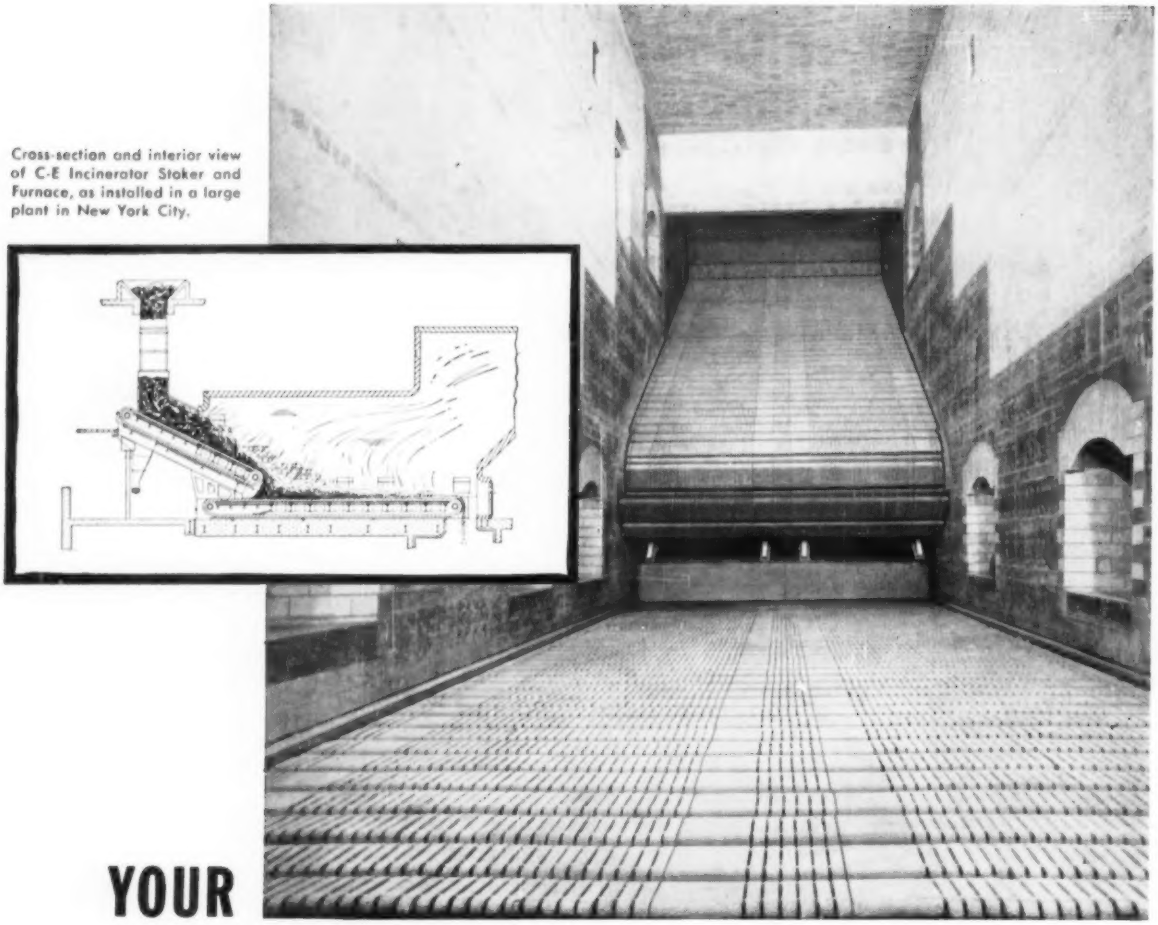
Power for Peaking

Steam Power Plant Clinic

American Power Conference

Economic Condenser Sizing

Cross-section and interior view of C-E Incinerator Stoker and Furnace, as installed in a large plant in New York City.



Combustion

DEVOTED TO THE ADVANCEMENT OF STEAM PLANT DESIGN AND OPERATION

volume 31 number 11 **May 1960**

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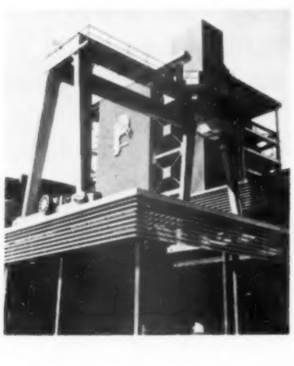
Striking use of modern architectural lines for outdoor power plants feature new City of Burbank, Calif. plant.



any way you look at it...

YARWAY REMOTE LIQUID LEVEL INDICATOR GIVES BRIGHT, INSTANT, ACCURATE READINGS

On its brilliant "wide vision" dial you can clearly read the level from any point in a 180° arc—and from a considerable distance.

Remote readings of levels in boilers, feed water heaters and other heat exchangers are instant and accurate because indicator operating mechanism is actuated by the varying head of the liquid itself, yet the pointer mechanism is never under pressure.

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A record of nearly 13,000 successful installations speaks for itself.

For details on construction, operation, installation and typical hook-ups, write for Yarway Bulletin RI-1825.

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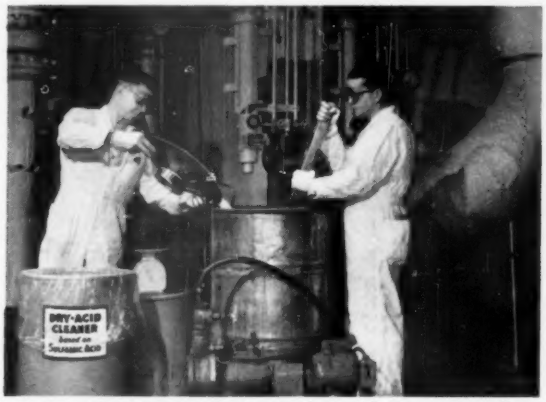
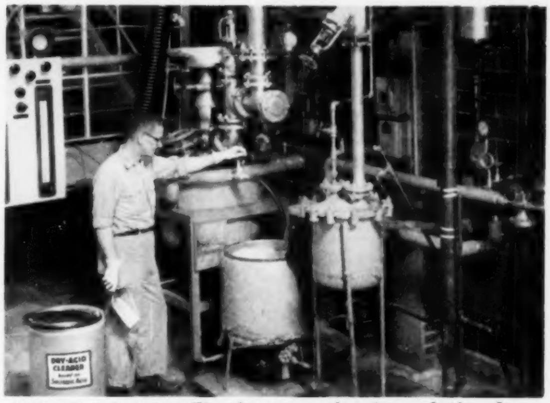
YARWAY
A MARK OF QUALITY IN STEAM ENGINEERING

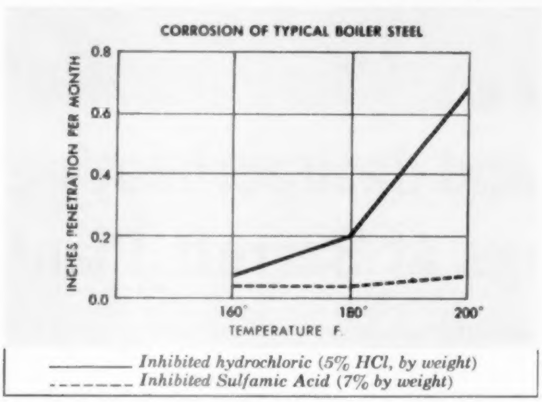
Descale boilers, heat exchangers, condensers rapidly, thoroughly with safer **dry acid** cleaners based on **Du Pont Sulfamic Acid**



SAFER TO HANDLE. Just pour dry acid cleaner based on Du Pont Sulfamic Acid from easy-to-handle, fiber drum into make-up tank. No danger from broken bottles, liquid spillage or corrosive fumes with these dry powders.



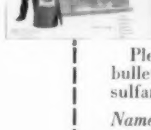
EASIER TO USE. Simply pump cleaning solution from make-up tank into boiler. Often dry acid cleaners can be added directly to equipment being cleaned. No elaborate equipment or specially trained labor needed.



LESS CORROSIVE. Sulfamic cleaners approach the strength of hydrochloric acid, yet are far less corrosive. Excellent inhibitor stability allows their use at higher temperatures for faster scale removal.

NON-FUMING. Sulfamic cleaners produce no corrosive fumes, either dry or in solution. Corrosion in vapor spaces of equipment during cleaning operation is eliminated, along with fume nuisance to personnel.

Ask your supplier for safer, effective cleaners based on Du Pont Sulfamic Acid, or mail coupon for additional information and names of formulators who offer these compounds.



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Please send me ☐ sulfamic acid general equipment cleaning bulletin; ☐ names of formulators offering cleaners based on sulfamic acid.

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Company ______________________________
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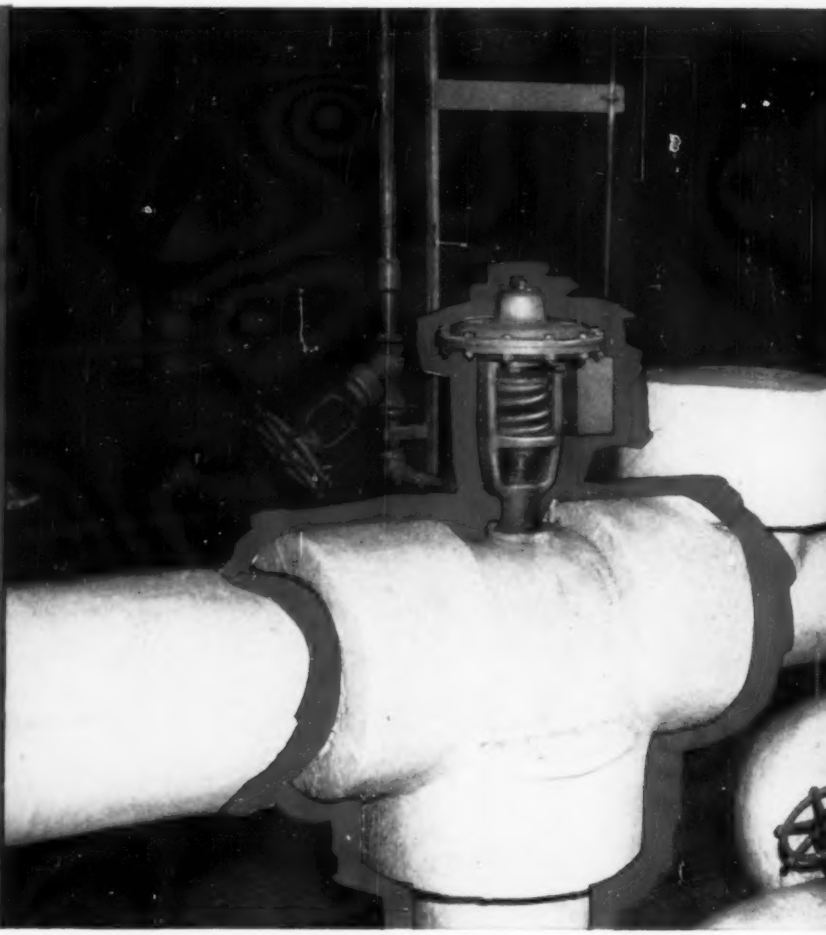
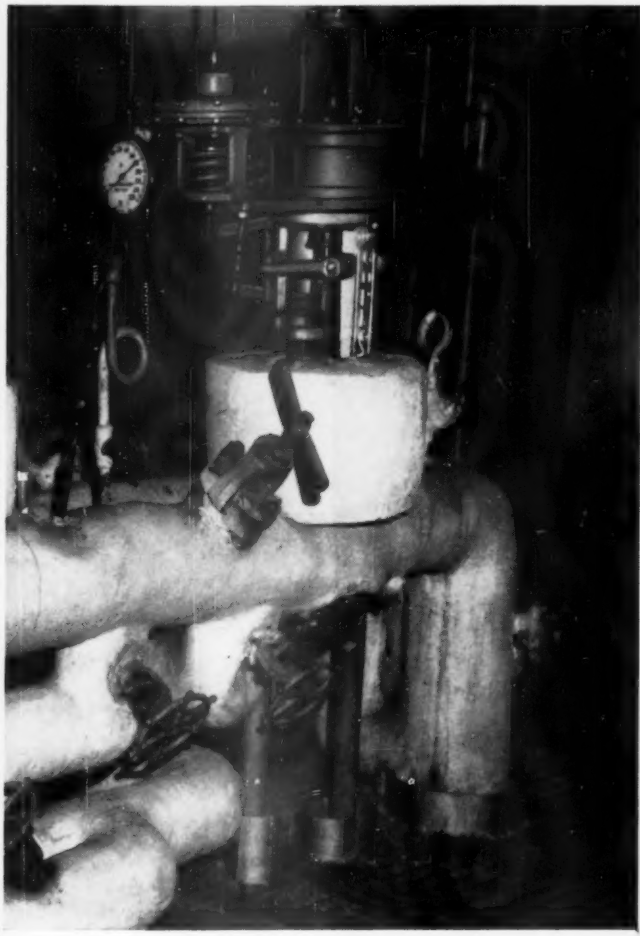
BETTER THINGS FOR BETTER LIVING ...THROUGH CHEMISTRY

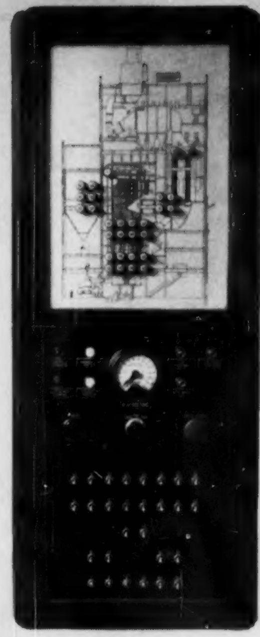
C-V NEWS NOTES



Above—**3,000 TO 17,000 POUNDS PER HOUR IN 30 SECONDS . . .** That's the load swing handled by the Copes-Vulcan pressure reducing and desuperheating station during winter nights when steam coils are put into service.

Upper left—**CARBURETOR-TYPE DESUPERHEATER USED AT BARRETT STATION.** Superheated steam is directed to pass the spray nozzle, which injects cooling water into the steam before it enters the mixing chamber. The flow of cooling-water is controlled at the spray nozzle to maintain a pressure drop across the nozzle at all flows. Cooling-water spray entering the mixing chamber is instantly evaporated, thus lowering the steam temperature to that desired. Final temperature may be held within plus-or-minus 5°F.

Available in standard 2" through 12" sizes, in 150- through 600-pound pressure standards for cast steel. Larger sizes and higher pressure standards are available on special application. Write for Bulletin 1056.



BARRETT STATION CONTROLS SOOT BLOWING on Unit 1 with a Vulcan Automatic-Sequential System. Vulcan Automatic-Sequential Control Systems can use steam, air, or a combination of both as the blowing medium.

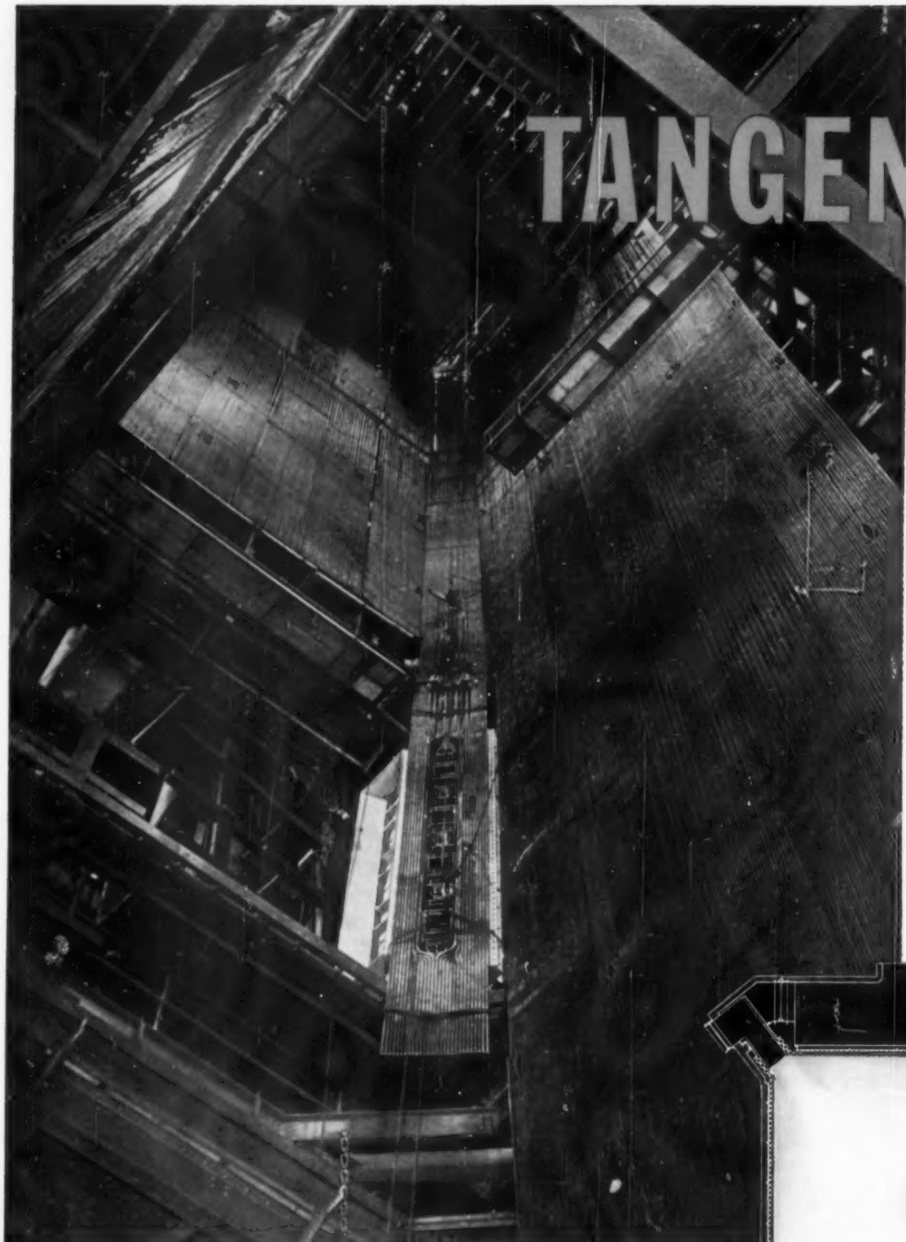
Without leaving the panel, the operator can pre-select any number of units to be operated through automatic sequence, or individually . . . may reverse any soot blower at any point in its operating cycle . . . or switch from automatic-sequential to single-unit operation.

For details on Vulcan's complete line of automatic soot blowing stations, write for Bulletin 1029. (Panel shown gives automatic-sequential control for up to 60 soot blowers.)

PISTON-TYPE VALVES FOR PRECISION CONTROL of pressure, temperature, and liquid level. Piston actuation develops high forces to deal with static and dynamic unbalances, and with stuffing box friction. These valves meet rigid requirements for unusually high operating force and accurate positioning.

Copes-Vulcan also builds diaphragm-type CV-D Valves which may be direct- or reverse-acting. They offer excellent rangeability, and can be profitably applied to many jobs.

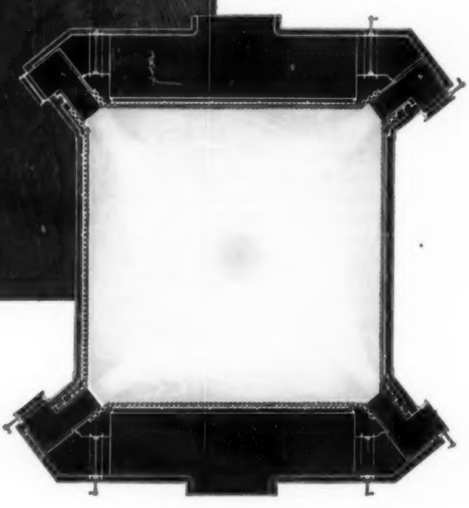
Both types available for pressure standards from 125 through 2500 pounds. Each valve is tailored to its job. For details, write for Bulletin 1027.

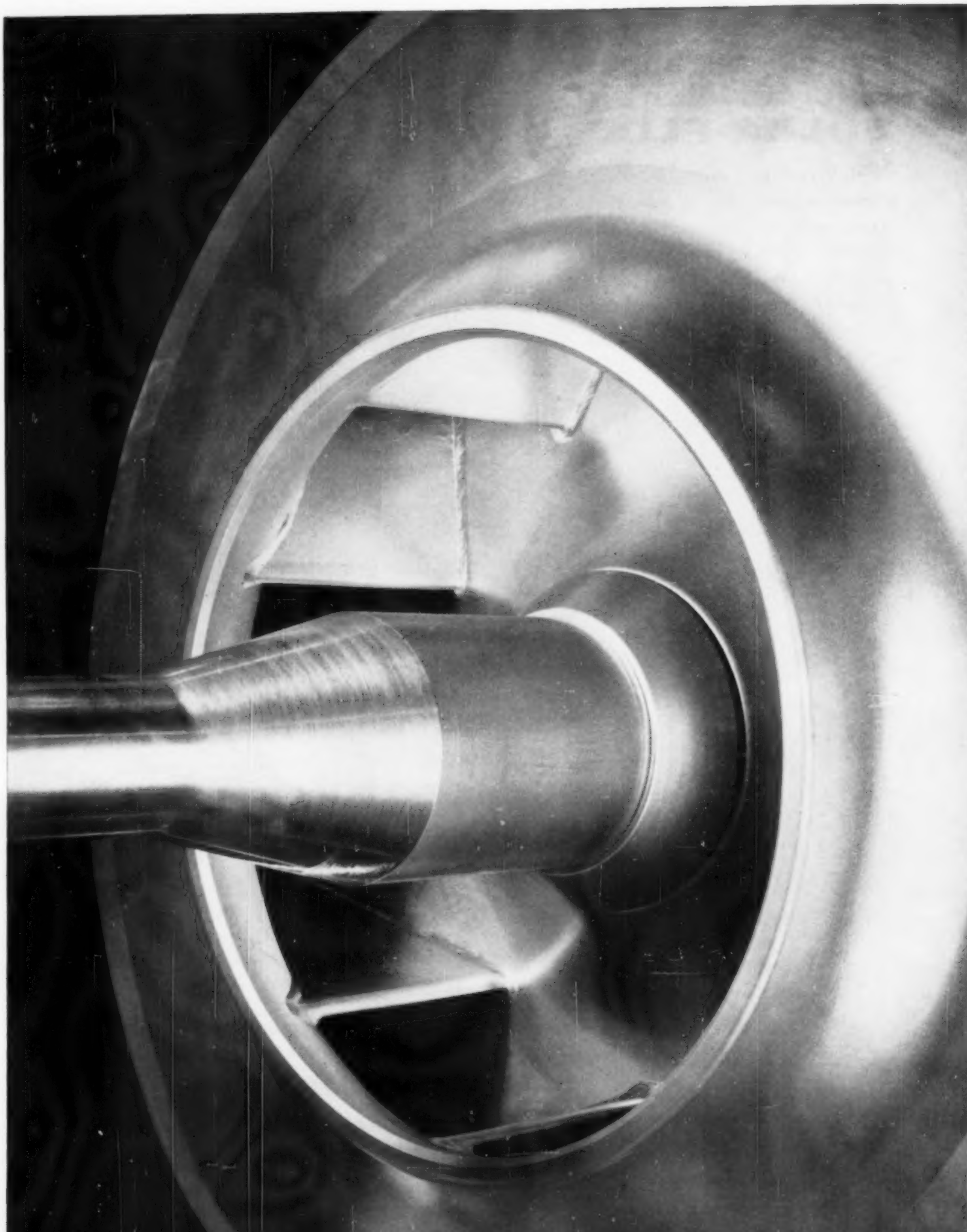


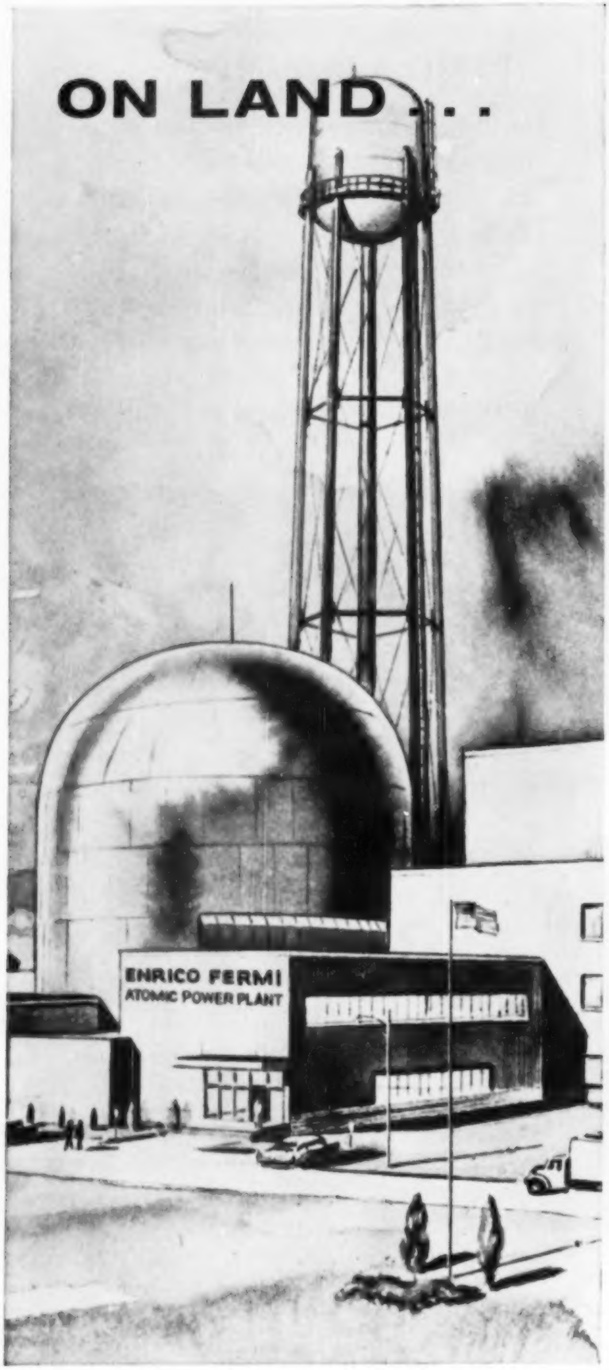
TANGENTIAL

Looking up into the furnace of a C-E Controlled Circulation Boiler. One of its four Tilting, Tangential Burners is being moved into position. The burner and its adjacent water wall tubes are shop assembled into a single panel and handled as one piece. All four corners of the furnace are similarly equipped. Tangential burners may be arranged to burn coal, oil or gas separately or in combination.

Sectional plan view through tangentially fired furnace showing impinging action of fuel streams from the burners which results in extreme turbulence and thorough mixing of fuel and air.







At the Enrico Fermi Atomic Power Plant, instruments and controls for both the "fast neutron breeder" reactor and the steam plant which it will "fire" are being furnished by Bailey.

Many of the new ships such as this super-carrier, USS Ranger, operate their boilers by Bailey Meter Control. Cargo ships, tankers, and passenger liners as well as Naval ships improve the economy and safety of their steam plants thru use of Bailey controls.

Hall Industrial Water Report

VOLUME 8 MAY 1960 NUMBER 3

394 Δισεκατομμύριον

The Greek word is billion. But you don't have to savvy Greek to know that 394,000,000,000 gallons is a lot of water. U. S. Industry is expected to be using this much each day in 1980.

Some of the water will be used as is. However, a sizable portion will require treatment to make it suitable for use. This might involve clarification, softening, silica removal, alkalinity adjustment, demineralization, or chemical conditioning to control corrosion or the accumulation of slime. And waste water may have to be treated for disposal or to recover valuable raw materials.

Water problems are Hall Laboratories' business. Experienced Hall engineers are ready to help you plan how to treat your water and how to tackle your water problems.

Excessive Sewer Tax

The size of his sewage bill shocked a manufacturer into the realization that he was contaminating a surface water supply. Waste water was being discharged in part to the municipal sanitary sewage system, in part to a small creek. Discharge to the creek was supposedly nothing but uncontaminated cooling water. However, the municipal authority proved that the water going to the creek was high in suspended solids and oil.

The municipal ordinance required the payment of a surtax if a plant's industrial waste waters contained suspended solids and BOD in excess of normal concentrations in sanitary sewage. Furthermore, plants discharging polluted waters to a surface supply had to pay sewer tax and surtax just as if it were going into the municipal sewers.

A lot of money was involved. Hall Laboratories was retained to determine how the contaminants got into the water going to the creek and how to stop them. After location of the sources, waste waters could be diverted to the sanitary sewage system. The preferred solution, however, was to clean up the waste waters, if this could be done economically. The latter solution would obviate payment of sewer tax and surtax on the water involved.

Hall staff engineer E. G. Paulson worked out procedures for disposing of the industrial wastes which made it possible to drop taxes from $600.00 to $75.00 monthly.

Aluminum Interference in Dyeing

Processing and dyeing of cotton fabrics had almost stopped in a Southern textile mill because of serious trouble in the dyeing operations. Practically all output was unsatisfactory because of migration of dye and deeply colored specks.

Hall industry specialist, Eric Laurin, was given a rush call. He soon established that the difficulty was due to the presence of an excessive amount of aluminum in the clarified and filtered process water.

Inspection of the filter plant showed the settling basins to be old and under capacity for the amount of water required. Floc formation with aluminum sulfate was poor and the water was carrying it out of the settling basins onto and through the filters.

By rearranging baffles in the settling basins and adjusting chemical feed, Laurin was able to improve the condition of the clarified water sufficiently to permit operation. However, floc was still carrying over to the filters and there was continuous danger of further trouble.

The next step was use of a Hagan Coagulant Aid. Results were immediate. Settling of floc in the basins was so rapid that there was practically no carryover to the filters. This was true even with 25% less aluminum sulfate. Several years have now gone by without recurrence of the costly condition.

Calcium Chloride Regeneration

When a dealkalizer in a hospital boiler plant suddenly stopped dealkalizing, Hall field engineer J. Printz received a rush call for help.

The sudden loss of effectiveness led Printz to suspect that the anion exchange material had become coated with something that was acting as a barrier between the water and the resin. Since the raw water seemed normal, his investigations took him almost immediately to the alkaline salt solution used for regeneration. He needed to go no further. One of the operators, to whom brine was brine, had dumped into the sodium chloride brine tank six hundred pounds of calcium chloride intended for preparation of brine for a refrigeration system. When the solution, with some caustic soda added, was used for regeneration of the anion exchanger, enough calcium compounds were precipitated on the exchange material to effectively insulate it from contact with water during operation.

Printz reasoned that if his diagnosis was correct the normal capacity of the dealkalizer should be restored by treatment with a solution of sodium hexametaphosphate (Hagan Phosphate), utilizing the ability of this material to sequester calcium and dissolve many calcium compounds. The procedure worked successfully and the equipment was promptly back in service.

Water is your industry's most important raw material. Use it wisely.

Industrial Water Problems Require Special Handling

There are no "stock answers" to industrial water problems. For information on how the Hall System can help you solve your particular water problems, write, wire or call address below.

HALL LABORATORIES

HAGAN CENTER, PITTSBURGH 30, PA.

Consultants on Procurement, Treatment, Use and Disposal of Industrial Water

DIVISION OF **HAGAN** CHEMICALS & CONTROLS, INC.

Here are America's first six CE-Sulzer once-through boilers: All have L&N Direct Energy Balance Control.

Company	*Station*	*Operating Pressure (psig)*	*Capability*
Dayton Power & Light Co.	Tait 4	2400	150 MW
Metropolitan Edison Co.	Portland 1	2400	165 MW
Dayton Power & Light Co.	Tait 5	2400	150 MW
Philadelphia Electric Co.	Eddystone 1	5000	325 MW
Cleveland Electric Illuminating Co.	Avon 8	3500	250 MW
Philadelphia Electric Co.	Eddystone 2	3500	325 MW

New L&N Direct Energy Balance Control Coordinates Once-Through Boiler and Turbine

Cleveland Electric Illuminating Company's Avon Power Plant is operating America's second CE-Sulzer supercritical unit (3500 psig). It employs the Direct Energy Balance System to control the boiler and turbine as an integral unit. Boiler-turbine output is changed in a pre-determined, orderly manner at the desired rate of generation change set by the operator, and output is kept within the capabilities of the equipment in service. From combined steam pressure and generation intelligence, D-E-B controls both fuel input and turbine governor valves. Excess air is "trimmed" continuously and automatically by flue gas oxygen analyzing equipment.

In designing the Direct Energy Balance method, L&N engineers sought a basic improvement over conventional combustion controls. Based on 30 years' experience in power plant measurements and controls, this new method was developed, subjected to simulation studies, and extensively field-tested. It will soon guide the operation of all six of America's CE-Sulzer once-through boilers. For information on this new concept in combustion control, call your nearby Field Office, or write for Reprint 463(8) to 4972 Stenton Avenue, Philadelphia 44, Pa.

Pioneers in Precision **LEEDS & NORTHRUP**
Instruments • Automatic Controls • Furnaces

Ljungstrom rotor being installed for Southern California Edison plant at Huntington Beach, Cal. Eight such Ljungstroms will serve four boilers at this station.

AIR PREHEATER SERVICE WILL STILL BE IN EFFECT ON THESE UNITS IN THE YEAR 2000

These new Ljungstrom® Air Preheaters, being installed at Southern California Edison's Huntington Beach Station, will be protected by an unusual service policy, one that guarantees regular inspection by Air Preheater engineers throughout the life of each unit.

There's no terminal date on this service – it's in effect as long as the units are in operation. In fact, Lifetime Air Preheater Service is still helping to keep the first Ljungstrom installation in the U.S. running perfectly, although it's been in operation almost 40 years.

Air Preheater provides first-rate *factory* service too, In one instance, in response to a last-minute decision to replace cold-end elements near the end of a scheduled shutdown, Air Preheater *fabricated and shipped* over 13,000 pounds of heating elements the day after the order was received.

Regular inspection and fast factory service are just two of the advantages Air Preheater offers its customers. Another is expert knowledge of boiler and preheater problems, gained through 35 years' experience. This combination—knowledge of customer problems and a *continuing* interest in them – probably explains why nine out of ten preheaters sold today are Ljungstroms.

THE AIR PREHEATER CORPORATION

60 East 42nd Street, New York 17, N. Y.

AT ASHLAND, KY. WORKS OF ARMCO STEEL CORPORATION

"Increased boiler capacity can often be obtained by modernizing boiler cleaning equipment. Another benefit of such modernization is more efficient utilization of the fuel . . . getting more heat into the steam for useful work and wasting less heat up the stack.

For example, at the Ashland, Kentucky Works of the ARMCO Steel Corporation there are four boilers that were unable to supply the growing steam requirements of the plant. The high exit gas temperatures suggested that a study be made to determine whether the cleaning could be improved to provide additional capacity. This study indicated that more steam from the same fuel could be expected if high pressure long retractable blowers were used for cleaning instead of the rotary blowers with which the boilers were originally equipped.

The expected results seemed sufficiently promising and it was decided to modernize the cleaning equipment of one boiler. The seven rotary blowers were replaced with four Diamond Long Retracting Blowers, one of which is shown below. This modernization proved to be justified as the boiler's maximum steam output was increased 11% and the exit gas temperature was reduced approximately 100° F.

A "Boiler Cleaning Modernization Program" is well worth careful consideration because it can mean substantial savings in so many ways. In addition to increased capacity and more efficient fuel utilization, there is reduced maintenance . . . also reduced operating costs when motorized units and automatic operation are installed. Even though your boiler cleaning was the best at the time it was installed, improvements since then will probably pay off. For many years Diamond has been doing continuous research to improve boiler cleaning and boiler cleaning equipment.

Ask the nearest Diamond office (or write directly to Lancaster) to make a study of your boiler cleaning . . . the possible savings may surprise you."

DIAMOND POWER SPECIALTY CORPORATION

LANCASTER, OHIO

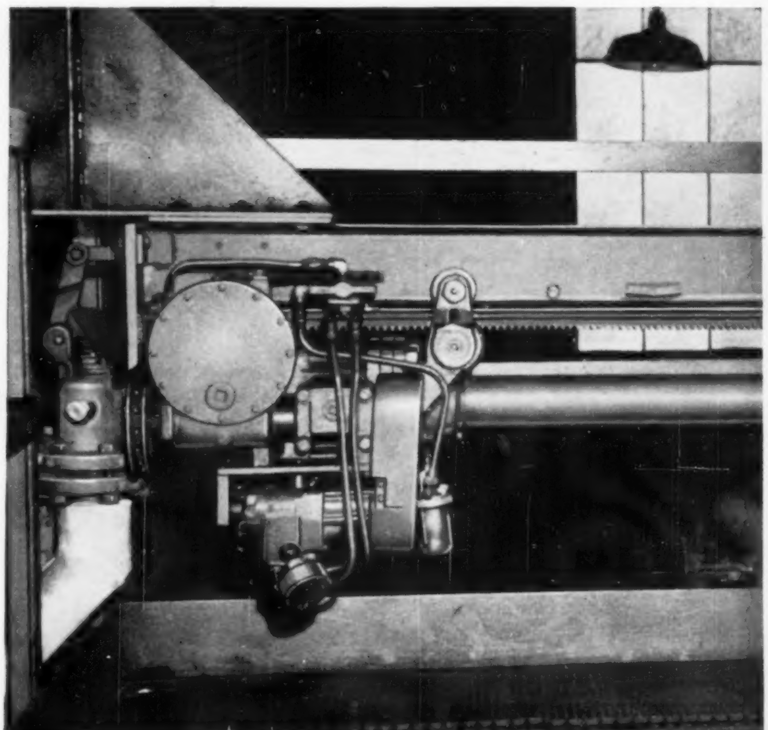
DIAMOND SPECIALTY LIMITED • WINDSOR, ONTARIO

Diamond

The Mark of

BETTER BOILER CLEANING

AT LOWER COST



UNRETOUCHED ACTION PHOTOS PROVE SUPERIOR PERFORMANCE OF

BUELL CYCLONES

an ordinary cyclone

THE BUELL CYCLONE
with Shave-Off

These are photographs of as true a test as can be made under laboratory conditions. Notice the difference in the dust patterns! In the Buell Cyclone, on the left, the dust at the top makes less than one revolution before it is trapped by the Shave-Off. It's then channeled down to the lower portion of the Cyclone, well below the clean gas outlet.

Why is the Buell Shave-Off so effective? Primarily because it harnesses the double-eddy current to convey the dust "fines" downward quickly, thereby promoting greatly increased efficiencies. In the ordinary cyclone, as shown on the right, these "fines" concentrate and recirculate at the top, causing erosion of the cyclone. To be collected, the fine dust must travel downward close to the clean gas outlet where much of it escapes. Buell Cyclones have made an impressive record in many years of trouble-free service. To see how their extra efficiency in the Shave-Off can pay off for you, send for our Cyclone Catalog #103. The Buell Engineering Co., Inc., 123 William Street, New York 38, New York. Northern Blower Division, 6404 Barberton Avenue, Cleveland, Ohio. (Subsidiary: Ambuco Ltd., London, England.)

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Combustion

EDITORIAL

Keeping the Wellhead Fed

An endless parade of press releases, news bulletins, progress reports march across the average editor's desk. Along the line of march you find differing views on almost any subject. One subject currently receiving considerable attention, probably because the school year is ending, is graduate study. We, as an industry, know (COMBUSTION, Feb. 1960, p. 25) that we need well equipped engineering graduates if our industry is to continue as a vigorous one. The differing views therefore, as regards our graduate schools, we find disturbing.

The growth of the graduate school in engineering has been tied closely to the growth in applied research within the university as contrasted to basic research due in large measure to World War II. Then, as one federal administrator puts it, "this changed philosophy toward research on the part of most colleges had come from the availability of governmental support, largely, though not wholly, the result of technological demands of our defense program. At the time when the colleges were having difficulty in finding enough money to keep their good faculty members, the availability of these research funds was of inestimable value. To a considerable extent, this is still the case."

But this growth has brought with it a disturbing situation. Dr. Fred H. Rhodes, former director of the school of chemical engineering at Cornell University, has this to say: "My greatest criticism of the American university education, in general, is the decreasing emphasis on really effective undergraduate instruction. There are several causes for this. One is the great number of sponsored research projects offered to and accepted by the universities. When a sponsored research program is offered to a university, the usual immediate effect is the withdrawal of the most competent professor in that field from some or all of his undergraduate teaching so that he may be assigned to the project."

Small wonder, then, that in the New York Times, March 1, datelined Princeton, N. J., appeared this lead paragraph to a story. "A convincing argument that a college education of high quality could be had at other than prestige colleges was offered today by the Woodrow Wilson National Fellowship Foundation." Apparently the smaller schools do not suffer from the attrition of research programs and are employing their best scholar-teachers toward student development rather than industrial product development.

We were much encouraged to see that the University of Illinois had devised a program of developing good engineering teachers, through graduate study, somewhat along the line of the Wilson Fellowship for general college teachers appealing enough to receive a Ford Foundation grant of $207,000—one of the first Ford Foundation grants in engineering and in sharp contrast to the $5,000,000 Ford Foundation grant for the long established Woodrow Wilson Fellowship. Programs such as the one at Illinois are vital to the development of our industry. It is only from such positive action that the industry's wellhead—adequately equipped engineering graduates—can be fed.

The utility industry has been keenly interested in the application of peaking power to its established capability. Much has been published. This paper brings together with generous use of illustrations much of the basic reasoning behind the role of peaking power and we publish it as an excellent primer for peaking.

*Power for Peaking**

By CLARENCE C. FRANCK, SR.†

Westinghouse Electric Corp.

THE problem associated with the total utilization of a power plant has always faced the electric utility industry. The tremendous capital investment required to engineer, construct, operate and maintain the power generating equipment, makes it necessary to utilize it to the utmost in order to obtain a reasonable return on the investment. When potential power generating equipment is idle, the overhead accumulates at a rapid pace. Practically all power generating equipment is projected on the basis that it will be unavoidably idle for a certain percentage of its useful life. In order to economically justify the project, this idle time is kept to an absolute minimum. Conditions surrounding power generating equipment expansion of the past has been based on the premise that the new equipment will be used to the utmost in at least the early stages of its useful life. Even on the assumption of a high use factor, power plant designers find it increasingly difficult to justify normal advances in operating conditions.

Characteristic trends in power generation of large utility systems have indicated that certain factors are influencing the fluctuation in the demand for power over periods of time. The ultimate result of this situation is to create a condition where at certain periods the demand for power is high while at other periods in the cycle, the demand for power has fallen off appreciably. The character of this demand has established a cyclic trend of crests and troughs and has created for the industry, the "peaking" problem.

These cyclic demands may manifest themselves in many forms and the most common of these will be briefly discussed.

1. Systems where a relatively high load is sustained for two thirds to one half of the daily period. Such plants must be designed for relatively high operating economy and will represent what might be considered the modern power plant of the present time. Such plants will normally be shut down after a certain hour and "controlled started" prior to the assumption of normal operation at another specified time.

2. Systems where several rather severe peaks would be encountered several times daily during the operating period. Such systems must be capable of being extended during these periods in order to satisfy the load demand.

3. Systems with seasonal peak loads. Such systems exist in areas with cold climates and the peaks develop during the low temperature periods of the year. Systems located in moderate temperature areas and, particularly those in extremely warm zones, have developed peaks as a result of the widespread application of air-conditioning.

4. Systems which might involve high load demands at some specific time throughout a period of years. Such systems might be located in areas where a large portion of the load consists of hydro-power. Should such a condition be compounded on an area which requires the pumping of water for irrigation, then the unavailability of hydro-power might be coincidental with a shortage of water and would necessitate that power be obtained for both basic use and pumping requirements.

The peaking problem has been well defined and numerous excellent papers have been written on the subject. The ultimate aim of those studying and prescribing for the problem is to obtain an economic balance between capital expenditures, operating hours and operating costs. The purpose of this paper will be to discuss certain possibilities and limitations in connection with the utilization of power generating equipment in practical thermal cycles for the large scale generation of electrical power.

Available Practical Cycles

In solving the problem posed by peak loads, the ultimate objective is to combine available apparatus to obtain the maximum power output from a minimum of installed equipment. In order to accomplish this end result, the power plant designer must make a careful survey of the equipment and combinations of equipment available.

The following practical cycles have been time tested

* Presented before the Southeastern Electric Exchange, April 8, 1960, New Orleans, La., and published by permission of the sponsoring organization.

† Consulting Engineer, Lester, Pa.

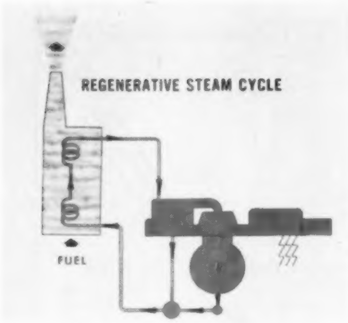


Fig. 1—Regenerative steam cycle

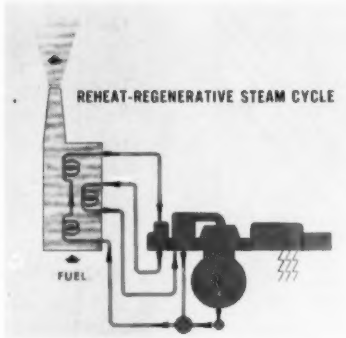


Fig. 2—Reheat-regenerative steam cycle

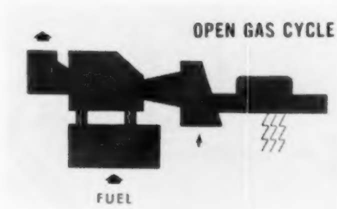


Fig. 3—Open gas cycle

for their suitability and reliability for the purpose of large scale production of electric power.

1. The regenerative steam cycle is shown in Fig. 1. In simple form, this cycle consists of a combustion chamber for burning the fuel, a steam generator, a steam superheater, a turbine, a generator, a condenser, a pump and a complement of feedwater heaters. This is a relatively simple thermal power cycle and has had widespread background of use in the utility industry.

2. The reheat-regenerative steam cycle is shown in Fig. 2. This cycle consists of the regenerative cycle with a reheat element added to the heat generating section and a corresponding change to the steam turbine to accommodate reheat.

This thermal cycle is now in widespread use and results in a power plant of relatively higher thermal efficiencies than the non-reheat cycle, but with increased capital expenditure and additional complications.

3. The open gas cycle is shown in Fig. 3. While this cycle is in its early stages of development and is limited to burning liquid or gaseous fuel, it presents great promise for one solution to the problem of peak loads.

The gas cycle represents the simplest form of power generation known to the power plant designer. Close observation of the gas cycle indicates that practically all of the essential elements of a power generating system are incorporated in a relatively few essential components.

The gas turbine which is the heart of the gas cycle has many advantages, particularly as applied to an element which might be subjected to severe cyclic operation.

There are additional available power generating cycles but none of these are of sufficient stature to match up to the expectations which is demanded by the electric utility industry.

Steam Power Cycles

From Figs. 1 and 2, it may be noted that while the steam turbine is only one link in the chain of steam power generation, it forms the keystone of the steam power generating cycle. It is a device which lends itself to a relatively precise proportioning to conform to a given established condition.

The steam turbine is a heat converting engine which is not greatly affected by the environment in which it operates. In order for the steam turbine to sustain a given power output, it is necessary to continuously supply energy. There is no means of building-in reserve or accumulating any appreciable quantity of energy in a steam turbine. Consequently, the mass of steam flowing through the turbine and the heat energy available, combine with the conversion efficiency of the steam turbine to generate a finite amount of power.

The turbine conforming to the non-reheat cycle, consists of a metering orifice at the inlet to the turbine and a corresponding metering orifice at the exhaust of the turbine. In the case of the turbine used with the reheat cycle, there is an additional orifice at the point where the reheated steam re-enters the steam turbine.

All the characteristics of a steam turbine are easily defined and it remains for the designer to provide sufficient area to admit steam and sufficient blade strength to satisfactorily resist the forces resulting from this steam passing through the stages of the turbine. The exhaust orifice of the steam turbine is somewhat more complex and a combination of steam flow, steam volume and exhaust annulus link together to form a natural balance at the point where the steam leaves the turbine and enters the steam condenser.

Fig. 4 shows a longitudinal view of the low pressure end turbine test facility which is located in the Engineering Experimental Laboratory at our Lester Works.

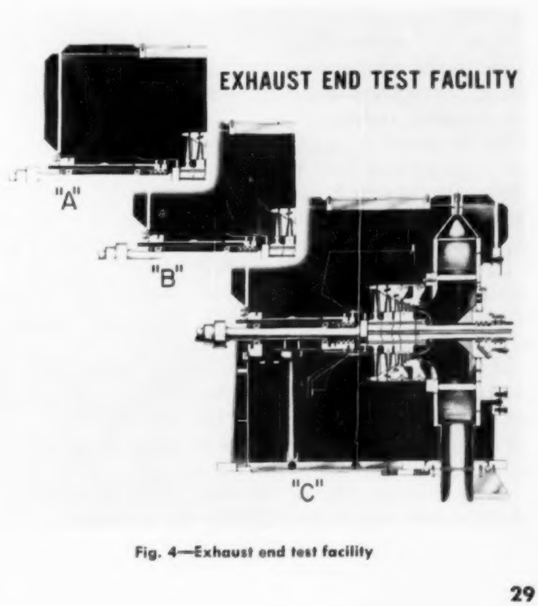


Fig. 4—Exhaust end test facility

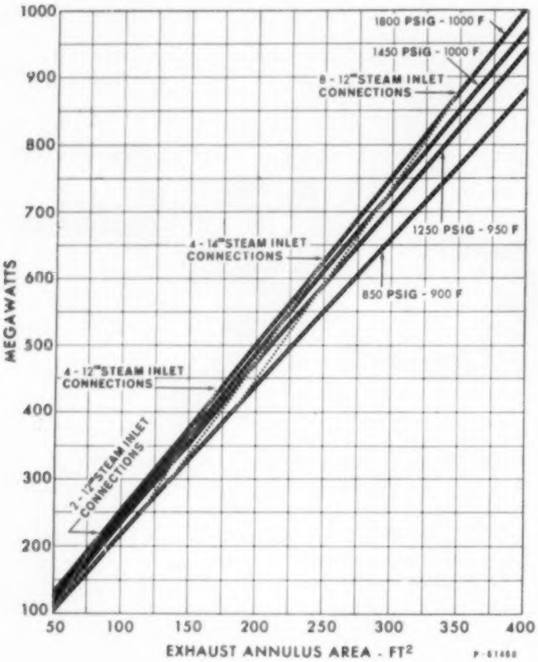


Fig. 5—Output from non-reheat steam cycle

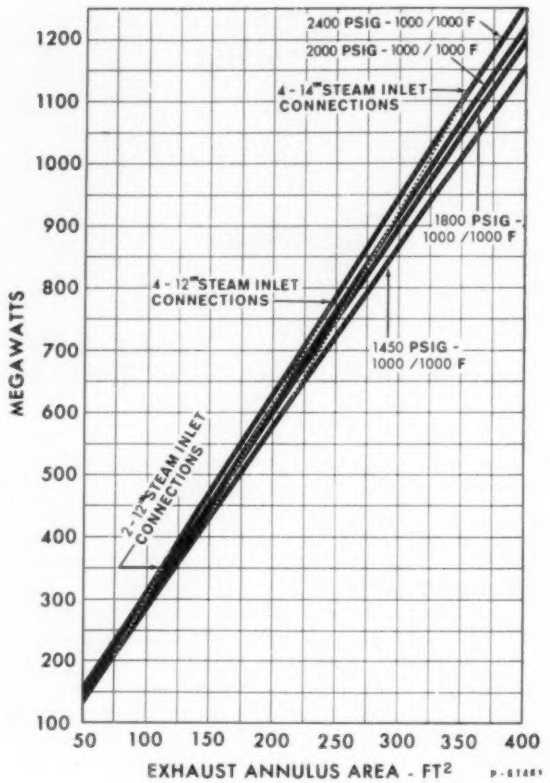


Fig. 6—Output from reheat steam cycle

This shows clearly the elements involved in establishing the orifice existing at the exhaust of the steam turbine.

View "A," at the top left, shows the low pressure end blading of a turbine exhausting directly into a housing, which is representative of the condenser. Obviously, in this situation, there is no restriction in the flow from the last row of blading and the area at this point forms the natural orifice which limits the exhaust flow.

View "B," center, shows a diffuser which has been added to the last row of blades in order to effectively guide the steam into the housing. The shape and proportions of this diffuser have been obtained by extensive air flow model tests. Such a procedure assures optimum shape and arrangement of the diffuser.

View "C," at the lower right, shows the exhaust hood added to the combination, and this completes the structure which represents the actual working parts of the LP end of a steam turbine.

By the application of the information obtained from this test facility, it is possible to calculate a pressure existing at the exit of the last row of blades based on a vacuum which is sustained by the condenser at the exhaust flange of the steam turbine. This relation in turn is predicated on a "Limit Exhaust Flow" through the turbine, which is established and determined by considerations of maximum allowable stresses in the blading.

The ultimate limitation in the load carrying ability of a steam turbine is determined by the flow through the last stages and the "Maximum Effective Vacuum."

Fig. 5 shows the limiting power output from a steam turbine with a given exhaust annulus area. This output is given as a function of the inlet steam pressures and temperatures when applied to the non-reheat cycle utilizing six stages of feedwater heating. Fig. 6 gives similar information for units operating on the reheat cycle. The dotted lines shown on Figs. 5 and 6 represent the nominal steam inlet pipe sizes required to pass the throttle flow which results in the "Limit Exhaust Flow" through the machine.

Having discussed the factors involved in limiting the predicted output of a given steam cycle, the associated equipment comprising the cycle will be given consideration.

(1) Generator

The generator must be provided with the necessary characteristics to permit the power developed by the steam turbine to be converted into electrical energy without excessive temperature or temperature rise

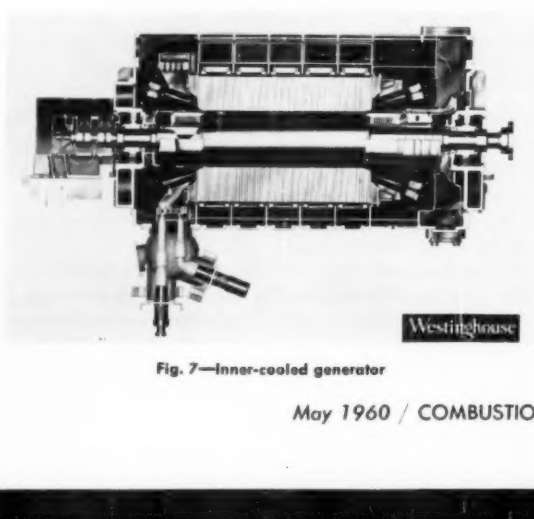


Fig. 7—Inner-cooled generator

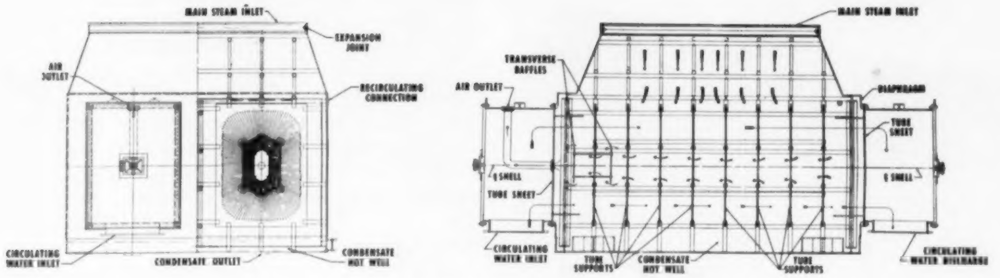


Fig. 8—Single pass steam condenser

occuring within the structure of the generator.

Fig. 7 is representative of an inner-cooled generator utilizing hydrogen as the cooling medium. Hydrogen-cooled generator load carrying ability is related to the hydrogen pressure.

It is recognized that generators can carry load beyond their rating and that some operators take advantage of this even though it involves operating at conditions other than shown on the capability curves recommended by the manufacturer This is done with recognition of the possible deleterious effect. Limitations involve not only the total temperature of the winding and the effect of temperature on the insulation but also the magnitude of the temperature differentials. In addition, one must consider the effect of higher loading on such components as the excitation system, bushings, interconnecting bus work and surrounding parts. These factors warrant detailed attention in the overall consideration of a peaking unit.

(2) Condenser

Fig. 8 shows the cross section through a typical single pass steam condenser. A steam condenser proportioned for peaking service must take into account the ability of the steam turbine to utilize the vacuum established by the condenser. Having established the Maximum Effective Vacuum which the steam turbine can utilize, the design of the condenser may be proportioned accordingly. Fig. 9 shows the "Minimum Required Surface" for "Maximum Effective Vacuum" versus steam turbine exhaust annulus with different cooling water temperatures. Such a steam condenser is proportioned to match the steam turbine and will have a resulting low vacuum at high steam flows and proportionately improved vacuum at lower flows through the exhaust of the steam turbine.

(3) Feedwater Heaters

It is interesting to note that when the steam flowing through the steam turbine results in the Limit Exhaust Flow, an increase in the final feedwater temperature from the feed heating system will result in an increase in the power output of the cycle.

This must be qualified by the number of feedwater heaters employed, and this characteristic is shown in Fig. 10. The information contained in Fig. 10 is representative of the characteristics of feedwater heating as applied to a peaking application for a reheat cycle. This curve is a typical example and each individual case must be based on its own specific conditions.

Fig. 11 is representative of feedwater heaters for high pressure and low pressure service.

(4) Steam Generator

The steam generator must be capable of supplying the steam in sufficient quantity and at the heat level required to permit the turbine to generate the specified output. The steam generator may be operated beyond its nominal rating, but such operation must be tempered with a knowledge of the consequences involved.

Steam Turbines for Peaking Service

Steam turbine for peaking service may be classified in several categories which allow the steam turbine to be adapted to the system characteristics. A description

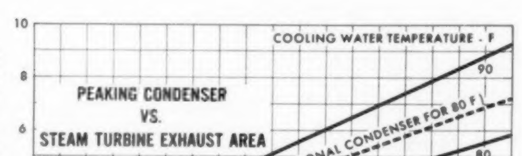


Fig. 9—Minimum required surface for maximum effective vacuum

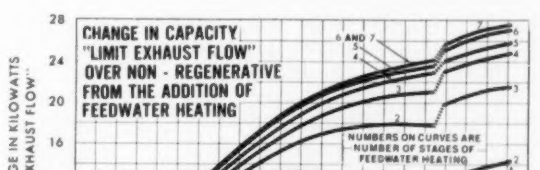


Fig. 10—Final feedwater temperature from feedheating system

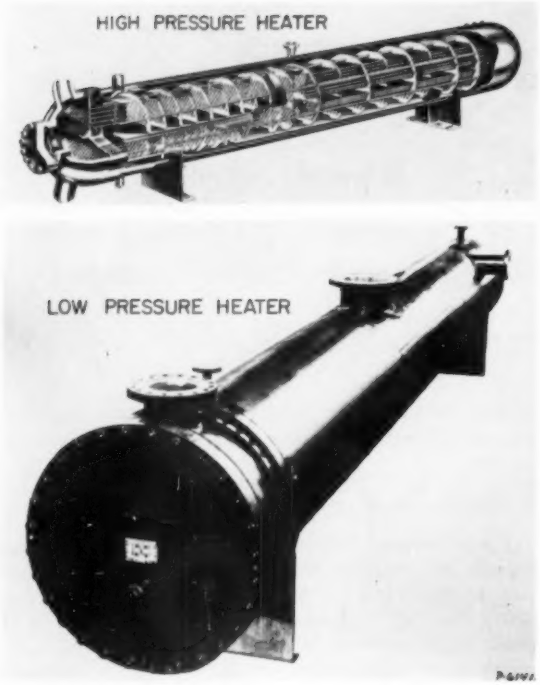


Fig. 11—Feedwater heaters for high pressure and low pressure service

of such steam turbines follows under separate heads.

Conventional Steam Turbines and Generators

These steam turbines are designed to operate on the reheat cycle with relatively high inlet steam pressures and temperatures. These steam turbines will have normally loaded exhaust ends and are expected to operate at relatively high thermal efficiency. Such plants are intended to be shut down during the "off peak hours" and "control started" in anticipation of the resumption of load. The equipment operating in conjunction with such plants is subjected to severe thermal cycling.

With the use of initial steam temperatures of 950 F, Westinghouse recognized that any transient thermal stresses could result in casing distortion and casing cracking. Cyclic operation, even though carefully controlled by conscientious operators, will impose temperature variations and thermal stresses on the turbine and generator structures. Such conditions were anticipated in the original design philosophy used in projecting Westinghouse turbines and generators for the past decade. Such features include adequate provision for differential axial expansion, transverse anchoring, separate stop valves, separate steam chests, flexible leads from the steam chest to the turbine casing, separate nozzle chambers, cold-worked copper, rotor end-turn construction, and Thermalastic insulation. Although the effects of cyclic operation will unquestionably increase the maintenance, the inclusion of such design characteristics will minimize the deterioration of the turbine and generator unit.

Fig. 12 is representative of the design features applied to the steam inlet section of a modern large steam turbine. This illustrates the application of steam cooling to reduce temperature gradients across the walls of the structure. The nozzle chambers are flexibly supported within the casing which permits them to expand independently of the inner case and of one another. The steam chests are separate from the turbine casing and connected by means of flexible pipes. The stop throttle valves are integral with the steam chests. An internal pilot valve in the stop throttle permits starting the steam turbine with the governor valves in the wide open position.

In addition to the design features as described above, the return of reheated steam is never brought directly into contact with the outer casing but is contained within a separate structure. This design protects the outer casing of the steam turbine structure from the normal large variations in reheat temperature.

Tandem Compound Double Flow 3600 Rpm Reheat Turbines

Fig. 13 is typical of this class of machines which have ratings from 50 to 225 Mw and are designed for the normal range of steam inlet conditions up to 2400 psig, 1050/1000 F.

The HP-IP element of this machine represents an advanced design which embodies features of mass flow cooling, rotor cooling and thrust balance under all conditions of steam flow. The use of the mass flow cooling principle reduces temperature gradient across, and the thermal stresses in, the walls of the high temperature zones of the inner casing.

Tandem Compound Triple Flow 3600 Rpm Reheat Turbines

Fig. 14 is typical of this class of machines which have ratings from 125 to 325 Mw. Such machines are used with steam inlet conditions up to 2400 psig 1100/1050 F.

The HP element of this machine is of the split opposed flow type and embodies the ultimate in component flexibility and minimization of thermal stresses resulting from thermal cycling. Steam from the HP element enters the IP element and flows to the crossover point. The IP element utilizes the mass flow cooling principle. One third of the steam from the IP element flows back across the IP inner casing and through the single flow LP ele-

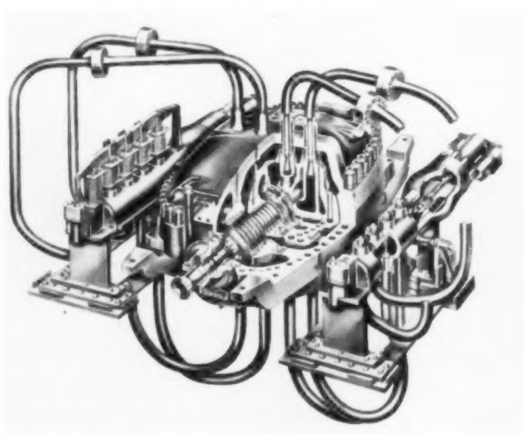


Fig. 12—Steam inlet section features

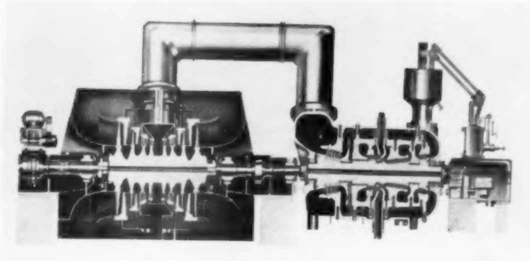


Fig. 13—Tandem compound double flow 3600 rpm reheat turbine

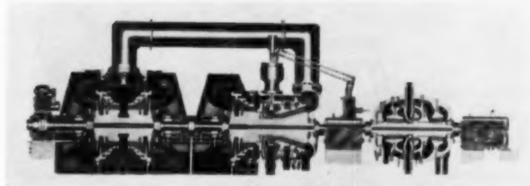


Fig. 14—Tandem compound triple flow 3600 rpm reheat turbine

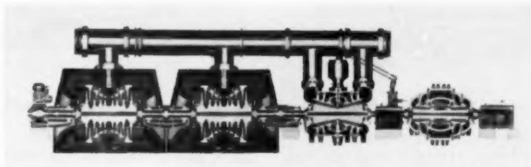


Fig. 15—Tandem compound quadruple flow 3600 rpm reheat turbine

ment to the condenser. The remaining two thirds of the steam flows through the exhaust openings in the forward end of the IP-LP cylinder and into the double flow LP turbine through two crossover pipes. The double flow LP elements have separate blade rings which are supported by the outer casing.

Tandem Compound Quadruple Flow 3600 Rpm Reheat Turbines

Fig. 15 is typical of this class of machines which have ratings of approximately 400 Mw. Such machines are used with steam inlet conditions of 2400 psig 1000/1000 F or higher.

Because of the large volumetric flow entering this machine, the HP element is of the double flow type.

An interesting feature of this type of machine is the arrangement of the two crossover pipes from the IP turbine to the two double flow LP turbines. This construction, which was verified by airflow model tests, reduces the complexity of the crossover piping system and minimizes the pressure drop through the pipe.

Cross Compound Quadruple Flow 3600/3600 Rpm Reheat Turbines

Fig. 16 is representative of this type of machine and it is composed of the same type of elements utilized in the tandem compound quadruple flow machines. In this case, arrangement is such that the HP element is connected to one of the double flow LP elements and the IP element is connected to the other double flow LP element.

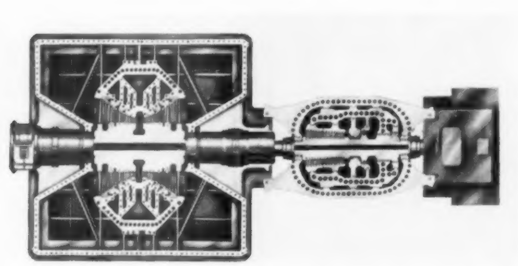
Cross Compound Double Flow 3600/1800 Rpm Reheat Turbines

Fig. 17 is representative of this type of machine which is used for ratings up to 500 Mw with steam conditions of 2400 psig 1000/1000 F or higher. The HP and IP elements are connected in tandem and operate at 3600 rpm. Such a machine makes available a large exhaust annulus which is effectively utilized in areas of extremely low temperature condenser circulating water. The double flow opposed type LP elements operate at 1800 rpm.

Cross Compound Quadruple Flow 3600/1800 Rpm Reheat Turbines

Fig. 18 is representative of this type of turbine which is used for very large capabilities.

This machine arrangement includes a double flow HP element and a double flow IP element connected in tandem and operating at 3600 rpm. The LP elements are connected in tandem and the steam exhausted from the machine flows into the condenser through two rectangular openings, one in the base of each turbine. The unit is supported by three transverse foundation members, one located at the forward end of the machine, the second



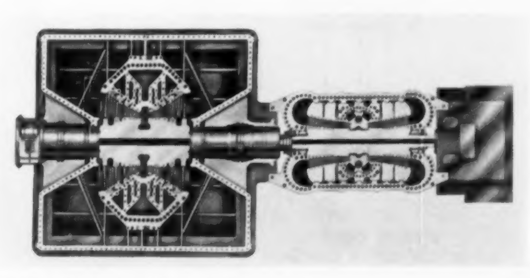
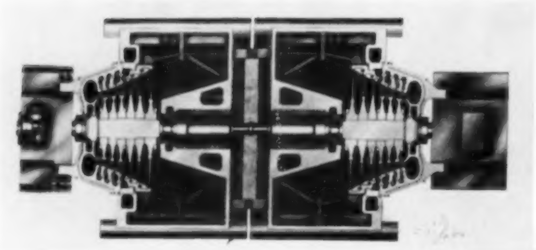


Fig. 16—Cross compound quadruple flow 3600/3600 rpm heat turbine



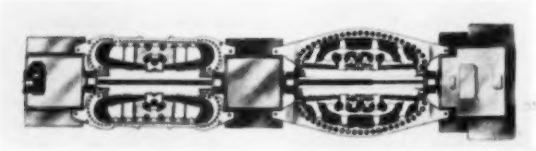
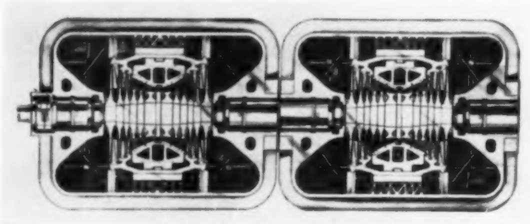


Fig. 17—Cross compound double flow 3600/1800 rpm reheat turbine



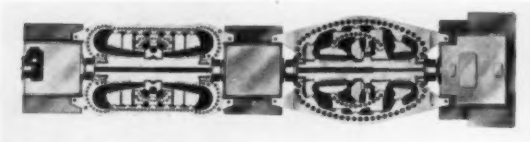


Fig. 18—Cross compound quadruple flow 3600/1800 rpm reheat turbine

at the aft end between the turbine structure and the turbine generator, and the third between the junction of the two low pressure casings.

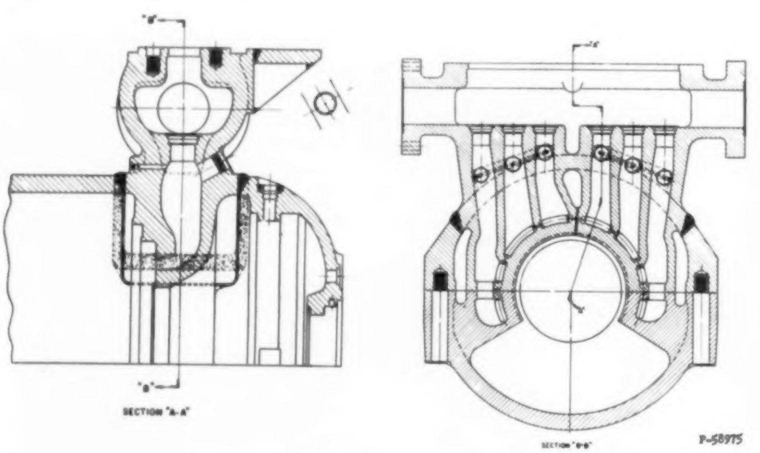
Low Pressure-Low Temperature (Minimum Operating Time) Steam Turbines

Steam turbines of this type will be designed for operation on the non-reheat cycle and the overall plant with which they operate will have inherently low thermal efficiency. In this design approach, the exhaust end of the turbine is heavily loaded in order to get the maximum power from the minimum structure. It is intended that plants of this nature operate relatively little time and are designed specifically for peaking service.

Plants for such intermittent service must be designed so that the equipment can be protected from deterioration during the long periods of inactivity. Although steam inlet pressures and temperatures for these plants are generally relatively low, the equipment is exposed to severe thermal cycling.

Although such units have separate stop-throttle valves, economics dictate that the steam chest and nozzle chambers be integral with a single wall HP casing. In order to obtain an approach to the flexibility afforded by separate nozzle chambers, a new concept in cylinder design has been developed. This offers some degree of flexibility without resorting to the expense and complexity of separate steam chests and separate nozzle chambers. The several views of Fig. 19 illustrates this construction. The steam chest and nozzle chambers are cast as a unit with slots separating groups of nozzle chambers, one from another. The cylinder proper is cast as a simple shell with a top opening to receive the nozzle chamber unit. The steam chest and nozzle chamber subassembly is then welded to the casing.

In addition to the design provisions described, a unit of this type should be given every possible consideration during its heating period prior to being synchronized and during the period when it is initially loaded. By using the pilot valve built into the stop-throttle, the unit may be heated and brought up to speed with all governor valves in the wide open position. The unit should then be synchronized on the stop valve with all the governor valves still in wide open position. The capacity of the pilot valve is proportioned to carry approximately 20% of the rated load. At this point, the



6-VALVE CHEST
ADMISSION COVER AND BASE

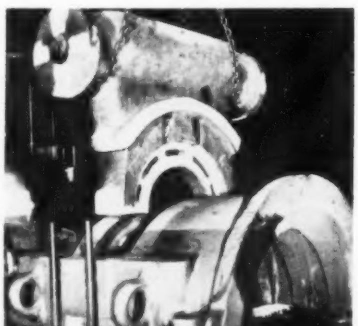


Fig. 19—Integral steam chests and nozzle chambers

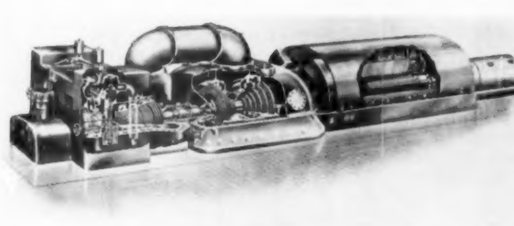


Fig. 20—Tandem compound double flow 3600 rpm non-reheat turbine

governor valves should be brought into service by adjusting the speed changer mechanism and the stop valve fully opened. This procedure will minimize thermal shock to the turbine casing and reduce nozzle partition wall temperature differentials.

Tandem Compound Double Flow 3600 Rpm Non-Reheat Turbines

Fig. 20 is a tandem compound double flow 3600 rpm non-reheat turbine and is representative of a unit for this class of service. Steam inlet conditions above 1250 psig 950 F would be difficult to justify economically. More common steam conditions for this type of service might be 850 psig 825 F.

Special Turbines for Peaking Service

An ideal application for an electric utility system having reasonably good base load characteristics, with relatively sharp peaks, might be in a steam turbine which was designed for optimum efficiency at relatively low loads with the inherent ability to carry higher loads at relatively lower efficiency.

Power plants and steam turbines for such service would require special consideration and involve special designs. A general discussion of steam turbines of this class and their potentialities will be given.

1. Such steam turbines might operate with relatively low steam inlet pressure and high steam inlet temperature and have a relatively large exhaust annulus for low leaving losses and high efficiency at normal rating.

By reducing the steam inlet temperature and raising the steam inlet pressure in such a way to maintain the same relative creep rate, large quantities of steam could be passed through the machine. This would result in increased power output at approximately the same heat rate depending upon the degree of change in temperature, pressure, boiler feed pump power, vacuum, and leaving loss.

The curve labeled ① on Fig. 21 shows the characteristic performance of such a steam turbine. It is to be noted that the partial load performance under the low pressure and high temperature steam inlet conditions is rather flat in nature.

2. In addition to the modifications as discussed under (1), such a unit might be designed with an extra stage of feedwater heating at the normal loads. The use of the heater with higher final feedwater temperature will improve light load performance. This heater may be bypassed at high loads to increase the capability of the steam turbine.

The performance characteristic of such an arrangement is shown by Curve ②.

3. Units designed to bypass additional heaters at maximum capability would have a characteristic similar to that shown by Curve ③.

4. Another approach to this problem would be to design the steam turbine to permit the induction of steam at an intermediate point in the turbine. In the case of a reheat steam turbine, this steam could be admitted at the hot reheat point.

Such an arrangement would provide for additional steam flow through the lower pressure stages of the steam turbine without increasing the physical size of the more expensive inlet features and main steam header.

The relative operating characteristics of such a machine are shown by Curve ④. It will be noted that the performance of such a steam turbine depreciates rapidly. By carrying this procedure to the Limit Exhaust Flow at Maximum Effective Vacuum, the capability would have the characteristics shown by 4A

5. A steam turbine can be designed to pass a throttle flow corresponding to the Limit Exhaust Flow. Fig. 21, Curve ⑤, shows the performance characteristics of such a unit. The exhaust flow would be the same as for

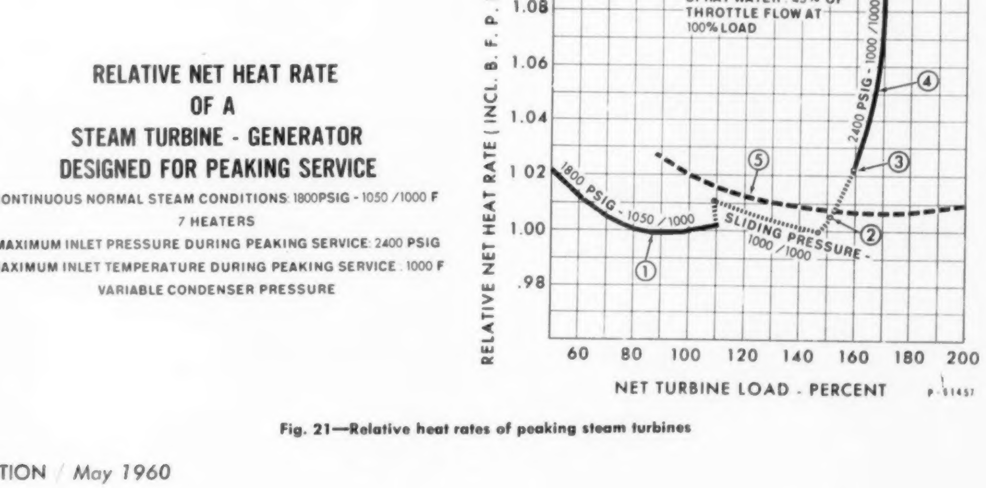


Fig. 21—Relative heat rates of peaking steam turbines

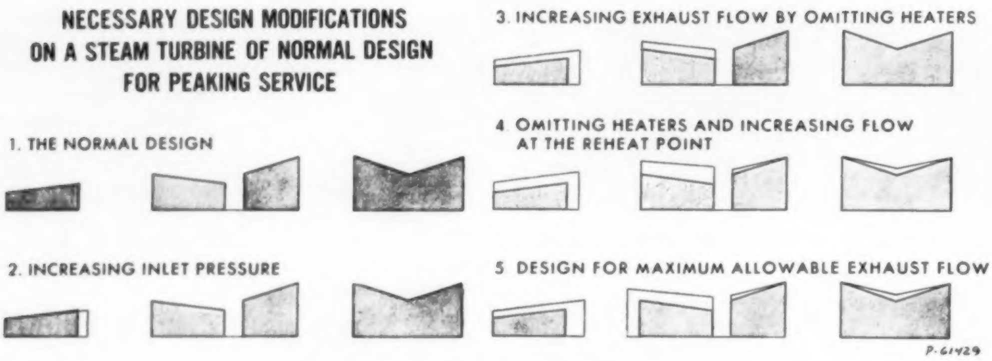


Fig. 22—Comparison of peaking steam turbines

the turbine utilizing the by-pass system under 4A.

The approximate capacity of the unit shown in Curve (5) of Fig. 21 is given by Fig. 6.

The characteristic blade paths of the machines discussed in Items 1 to 5 of Fig. 21 are shown in Fig. 22. While it must be understood that the scale of these illustrations is not exact, they are fairly representative of what must be done to the steam turbine to accomplish the end results as described in Items 1 to 5.

The provisions, which must be made by the turbine designer to accommodate the larger flows required to get the greater output, become apparent from this presentation.

An interesting by-product of the situation covered by Items 1 to 5 is shown in Fig. 22. This illustrates the vacuum characteristics of a condenser designed under the rules by which Fig. 9 was established. Consider a condenser with Minimum Required Surface for Maximum Effective Vacuum to be operated at other than design flow. Fig. 23 shows the resulting vacuum at percentages of design flow for different cooling water temperatures.

The breaks in the lines of constant cooling water temperature result from the application of one and two circulating pump operations. In order to get the most economical condenser for peaking service, the velocity through the condenser tubes is increased at maximum flow. By dropping out one circulating pump at flows below 75 per cent of maximum flow, the velocity through the tubes is reduced appreciably. In addition, the use of one circulating pump at light loads saves on the pumping power with attendant improvement in performance.

The performance as indicated by Fig. 21 incorporates this by utilizing improved vacuum at the lower flows.

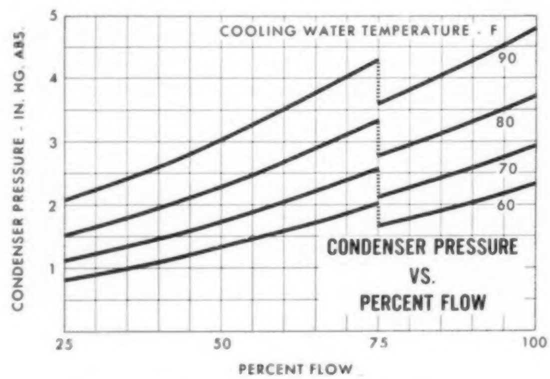


Fig. 23—Vacuum characteristics of peaking condensers

Steam Turbines for Boiler Feed Pump Drives

While Fig. 21 is expressed in relative net heat rates and includes an allowance for the difference in power required to pump the boiler feedwater, no reference has been made to the boiler feed pump drive. Fig. 24 shows two longitudinal sections of typical feed pump drives.

The upper section shows a typical non-automatic extraction, non-condensing turbine which may either take steam from the cold reheat point or from the main

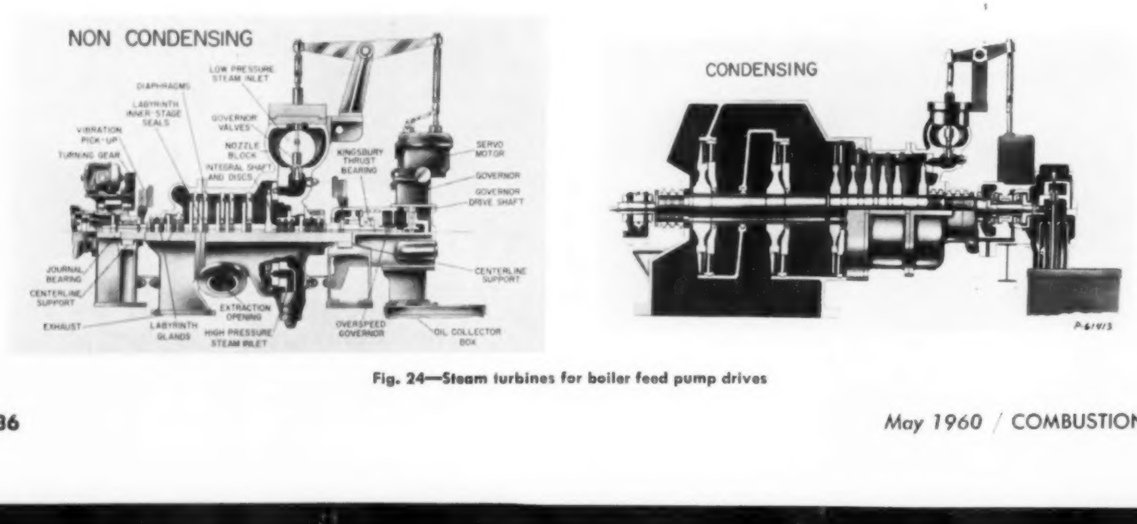


Fig. 24—Steam turbines for boiler feed pump drives

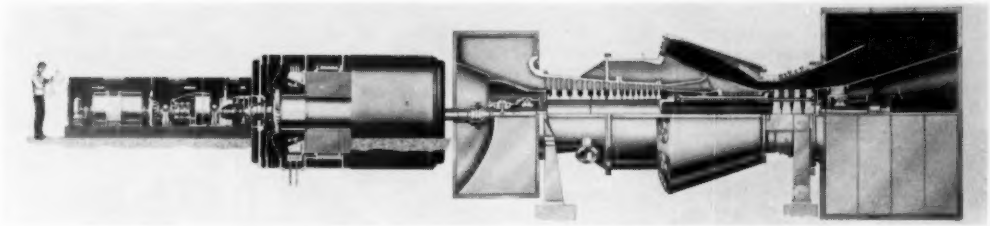


Fig. 25—22 mw peaking gas turbine-generator

steam header. This steam turbine exhausts into the lower stages of the main steam turbine.

The lower section shows a typical condensing turbine which exhausts into the main condenser. The advantage of this type of steam turbine lies in the fact that the additional exhaust annulus adds to the potential load carrying ability of the system.

The advantage of the steam turbine driven boiler feed pump is that it may be operated at variable speed over the load range to better conform to the pumping requirements of the system. Also, for the variable pressure requirements, the variable speed characteristics of the steam driven boiler feed pump is advantageous.

The application of the steam driven boiler feed pump usually necessitates the use of a motor-driven boiler feed pump for starting the plant. By taking advantage of the proper proportioning of this motor driven boiler feed pump, the steam driven boiler feed pump may be shut down at the lighter loads. The elimination of the steam driven boiler feed pump from the cycle under these conditions will improve the overall performance of the cycle.

It must be understood that the presentation shown in Fig. 21 is an approximation and each specific case must be worked out on its own merits.

Gas Turbines for Peaking Service

The gas turbine has excellent characteristics for application to peaking service, particularly where the fuel used is either gas or oil. Even for those cases where coal is the basic fuel, the utilization of a gas turbine to carry peak loads with either oil or gas may be economical. Fig. 25 is a cross section of a 22 Mw gas turbine showing the arrangement of the compressor, combustors, gas turbine, generator, exciter and starting auxiliaries. This unit is designed specifically for peaking service.

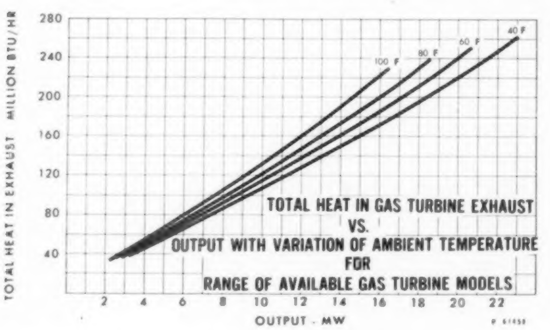


Fig. 26—Capabilities of gas turbines

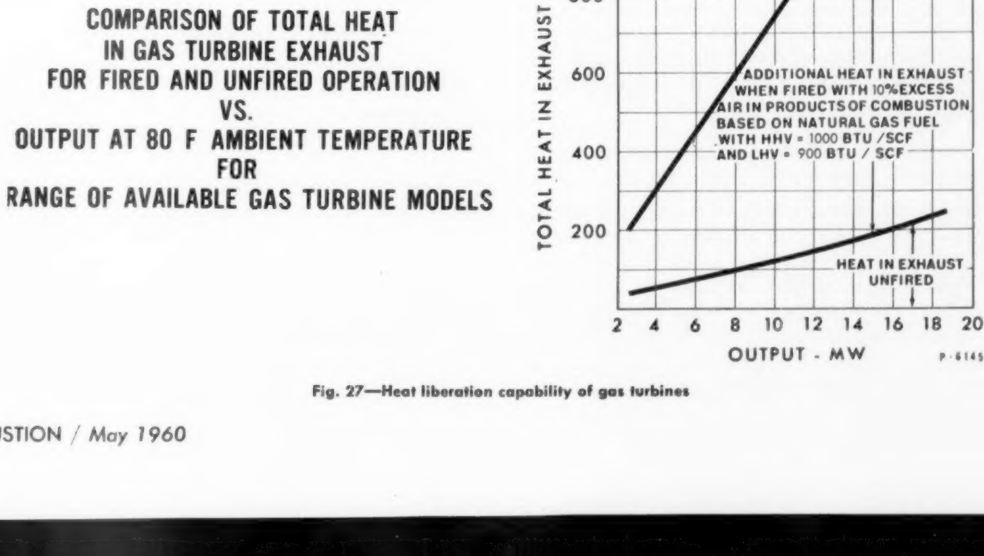


Fig. 27—Heat liberation capability of gas turbines

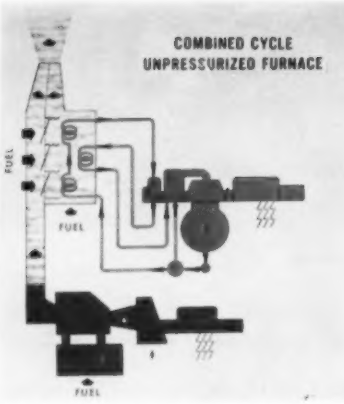


Fig. 28—Combined cycle (unpressurized furnace)

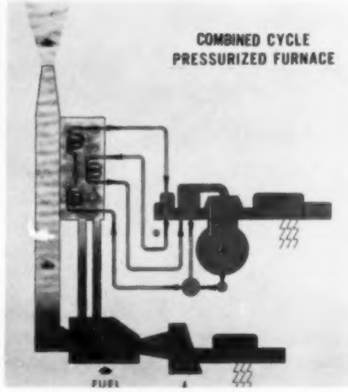


Fig. 29—Combined cycle (pressurized furnace)

Fig. 26 gives an indication of the range of capabilities possible with existing gas turbine frames. Any given point will depend upon the actual physical proportions of design of the specific gas turbine involved. The curve has been prepared to show the resultant heat in the exhaust of a conventional open cycle gas turbine. This information is presented as a function of ambient or inlet temperature of the air to the gas cycle.

Combined Steam and Gas Cycle for Peaking Service (Conventional Steam Generator)

One promising application for the solution of the peak load problem is the combination of the steam and gas turbines in the generation of power. This application involves the integration of the power generating equipment and the heat liberating equipment.

Fig. 26 suggests the possibility of large quantities of heat available as a by-product of the gas turbine cycle. It is recognized that in order to regulate the temperature of the hot gas to the gas turbine, it is necessary to use large amounts of "excess air." As a result of the use of air in tempering the gas, the exhaust from the gas turbine is high in oxygen.

Assuming that fuel is mixed with the normal exhaust of the open cycle gas turbine, it is possible to provide large quantities of heat liberation and, at the same time, utilize the heat which is in the exhaust gases of the gas turbine. Fig. 27 shows the potential heat liberation from the products of combustion with 10 per cent excess air and a natural gas fuel.

Fig. 28 shows a possible means of utilizing this additional heat release at specific points in the steam generator. This combination might be effective in providing for the special requirements of the special steam turbines for peaking service as illustrated in Fig. 21.

Combined Steam and Gas Cycle for Peaking Service (Pressurized Steam Generator)

The most effective use of the combined steam and gas cycle appears to be in the total integration of the equipment. In such an application, the gas side of the steam generator would serve as the combustion chamber for the gas turbine. Pressurization of the combustion chamber of the steam generator would tend to reduce the size of the steam generator for a given output. Also, increasing the degree of pressurization for the steam generator may result in an increase in the heat release to be used in the generation of steam during the periods of peak load operation.

A gas turbine using solid fuels is not considered as practical at the present time. Problems associated with the gas turbine, which must of necessity take the products of combustion from the steam generator, have not been solved. The effects of the fly ash on the gas turbine blades would result in questionable reliability. Chemicals in the combustion gases passing through the gas turbine would have a detrimental effect on the internal elements of the gas turbine.

While the status of the art discourages the use of solid fuels in the totally integrated steam and gas cycle, there appears to be no great obstacle in the use of natural gas or fuel oil. Fig. 29 would be typical of such a plant which might be used in an area where the normal fuel burned is either oil or gas.

The advantages of the pressurized combustion chamber of the combined steam and gas cycle might further be enhanced by the application of a "once through" principle to replace the conventional drum of the conventional steam boiler. With control of the pressure ratio at which the gas turbine operates, the heat release of the boiler might be increased within the confines of a given combustion chamber. With the "once through" principle of steam generation, the control of temperature and the choice of pressure would make it possible to utilize some of the steam turbine features previously discussed. These combinations might offer some possible solutions of the problem presented by peak loads.

Conclusions

The information presented in this paper should give power plant designers and operators a better understanding of the components comprising the steam power cycle.

There are misconceptions relative to the overall ability of equipment to perform beyond its design limitations. A better knowledge of the fundamental considerations should rationalize such misunderstandings.

These discussions provide a straightforward approach to the disclosure of available potentialities in solving the problem of "Peaking Power."

The cooperation of our generator designers in the East Pittsburgh Works is acknowledged.

By IGOR J. KARASSIK*

Worthington Corp.

The boiler feed pump and its associated equipment represent a major operating and maintenance consideration in today's power plant. Here we run in question and answer form a series of clinic sessions on various boiler feed pump problems. The replies are the work of one of the topmost pump authorities and give specific information which we hope will prove valuable to our readers.

Steam Power Plant Clinic—Part XVI

QUESTION

We have an installation of two 100 per cent capacity boiler feed pumps, one of which is on standby service. They are designed for 700,000 lbs per hr and 1200 psig discharge pressure. The minimum flow of these pumps is 100,000 lbs per hr. The minimum flow control valve opens when flow is reduced to 100,000 lbs per hr and stays open until the flow reaches slightly above 200,000 lbs/hr so as to avoid hunting. At that increased flow it will close suddenly. This imposes a severe step input to the control system and if the pumps are on hand control, requires immediate corrective action. Is the expense of proportioning the amount of recirculation not justified?

I would also like to know why we could not utilize the feedwater flow nozzle in the discharge piping after the heaters instead of installing a separate orifice in the pump suction or discharge line? (W.O.F.)

ANSWER

Unfortunately, there is no simple means to avoid this problem if an open-and-closed recirculation valve is used. Its existence has led in a few cases to the installation of modulating control valves which maintain the sum of the flow to the boiler and of the recirculation by-pass flow to the minimum specified by the pump manufacturer. The difficulty of this arrangement arises from the fact that the modulating valve is called upon to handle variable flows all the way from the full value of the minimum recirculation flow down to zero. Thus, the valve may be throttling off a pressure from a negligible 50 psi (when it is wide open) to a maximum of full shut-off net pressure of the pump (when it is almost completely closed off). This leads to a high rate of valve wear and a possible source of high maintenance expense. In the past, I have heard of modulating valves which would not last more than a few weeks on such severe service.

I have been told recently, however, that several valve manufacturers have developed new designs which are better able to stand up under this type of service. You may therefore wish to contact some of these manufacturers and inquire what may be available today.

One possible solution to reduce the shock of the full change of flow is to use two orifices in parallel, each equipped with its own open-or-closed by-pass valve and to provide sequential operation for these valves as the need arises. The shock to the system will be less severe, but the installation is somewhat more expensive.

If, as in your case, 100 per cent capacity pumps are involved, the feedwater flow nozzle in the discharge piping can certainly be used to actuate a minimum flow signal instead of installing separate flow orifices in the pump suction or discharge lines. There is, however, a distinct risk introduced whenever the standby pump is brought on the line to replace the pump which is running. If the transfer takes place while the demand of the boiler is in the low range and pump operation is taking place in the relatively flat part of the head-capacity curve, one of the two pumps could back the other pump off the line and the check valve in the discharge of this latter pump will close. This pump could suffer severe damage since the main flowmeter cannot distinguish from which pump feedwater is delivered to it.

Of course, there is a means available to avoid this danger. It would consist of a relay introduced into the operation of the individual by-pass valves which would maintain these valves *open* regardless of flow as long as both pump drivers are energized. This would not lead to any waste of power since the period of time during which both pumps are on the line is insignificant in the case of 100 per cent capacity pumps.

(Continued on the following page)

* Consulting Engineer and Manager of Planning, Harrison Div.

QUESTION

I have read a number of your articles on the effect of sudden main turbine load drops or trip-outs on the conditions prevailing at the suction of boiler feed pumps which operate in open feedwater cycles. You have developed a method for calculating both the allowable and actual rates of pressure decay in the deaerator from which the pumps take their suction. But in the final analysis, the question of whether circumstances are or are not favorable hinges on the behavior of the feedwater flow immediately following a drastic reduction in load. Is there any accurate and dependable means to predict this behavior?

ANSWER

There may be such means available, but unfortunately I have not yet been able to develop this. Instead, one is forced to resort to various approximations based primarily on past experience with boiler controls similar to those being provided for the unit under study. There are in addition, as you will presently see, certain general assumptions which can be made—assumptions which can be as pessimistic or as optimistic as the designer is inclined to be.

As a matter of fact, it was mainly because of the difficulties of evaluating the exact behavior of the feedwater flow following a sudden reduction in load that my colleagues and I developed a method of comparing *allowable* and *actual* rates of decay which postpone the determination of this behavior.* Thus, all the calculations of these two rates proceed independently of a decision as to the exact value of the feedwater flow *after* the load drop, and a curve is constructed similar to that shown on Fig. 1. After this, consideration is given to whether the maximum flow after load drop will or will not exceed the maximum safe value indicated by the intersection of the *allowable* and *actual* rates of pressure decay.

To begin with, we must realize that even when the feedwater system is provided with a two- or a three-element feedwater regulator, the feedwater flow will not instantaneously follow the steam flow as soon as the steam demand is reduced. Because of the time lag which exists between the reduction in boiler demand and that of the fuel burning rate, and because of the heat reserve in the steam generator, there is a momentary rise in the boiler pressure, with the resultant collapse of some of the steam and water bubbles in the boiler drum and a lowering of the boiler drum level. This reduction in level overrides to some degree the impulse from the change in steam flow. Therefore, there will generally be a definite lack of correlation between feedwater and steam flow following a sudden drop in load. The exact degree of the difference between these two flows will depend upon the particular type and setting of the feedwater controls and upon the characteristics of the regulator.

As an example of the variations which may occur, some feedwater regulator controls incorporate a bias in the effect of the drum level impulse upon the positioning of the feedwater regulator valve. By means of this bias, the boiler drum level at low loads is permitted to fall some six to twelve inches below the level at maximum load.

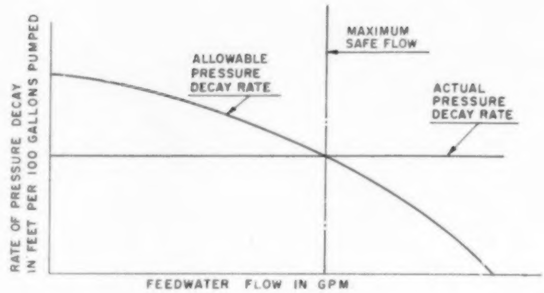


Fig. 1.—Comparison of pressure decay rates following a sudden, drastic reduction in pump load

This mitigates to some degree—but not entirely—the effect of boiler drum shrinkage after a load drop.

Until recently, I had only one completely documented case of sudden load drop on which I had full data on the behavior of the steam flow, feedwater flow and drum level. The test had been carried out on a 100,000 kw, 1450 psi throttle pressure unit and the turbine load had been reduced almost instantaneously from 117,500 kw to 28,000 kw. The recorded values of steam and feedwater flows and of drum level are reproduced on Fig. 2 This installation was provided with a bias in the feedwater regulator controls and had there not been a provision for such a bias, the sudden upswing in feedwater flow which took place after the first minute would have been even more violent and would have taken place earlier.

In my search for additional documented material, I contacted a number of my utility friends with a request for charts obtained after sudden load reductions. I must state that the response was immediate and most cooperative Unfortunately, two factors have united to prevent me from deriving accurate and dependable conclusions on the behavior of the feedwater flow:

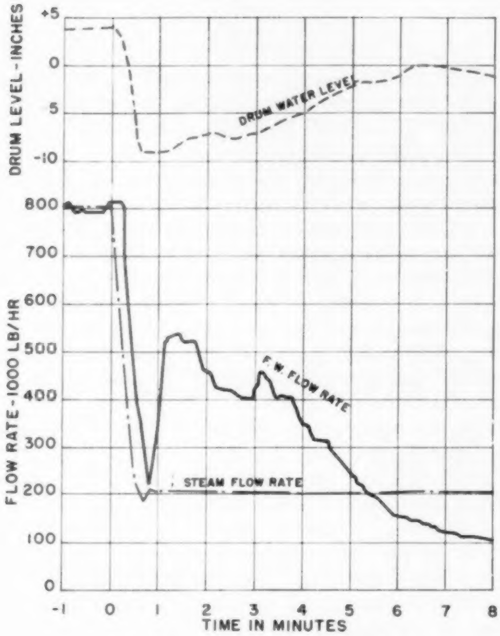
(1) The charts that were supplied to me were standard 24-hour charts and therefore flow behavior over a total duration of some two or three minutes is difficult to analyze. (I should mention that the data reproduced on Fig. 2 were obtained from a "souped-up" chart mechanism, with a 60 to 1 speed-up gear which transformed the usual circular chart into a 24-minute one).

(2) There was no correlation whatsoever in the charts I received with the data provided on the type and setting of the feedwater controls. In other words, there were almost as many variations in the behavior of the feedwater flow as there were documented cases.

For general interest, I have reproduced in Figs. 3 through 5, a group of three charts which are quite typical of the dozens which were supplied to me You will agree that it would be difficult to make any general rules with regards to the relation of feedwater flow to steam flow from these data.

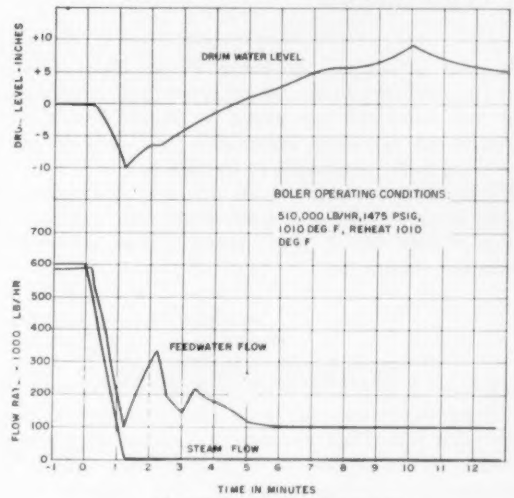
I mentioned earlier that certain general assumptions can be made with regard to this relation. If one were to be extremely pessimistic, one could assume that no reduction of feedwater flow whatsoever would take place for several minutes and, as a matter of fact, that the feedwater regulator valve would swing wide open. In this position, the feedwater flow would probably exceed the normal flow corresponding to the rated maximum turbine

* "Centrifugal Boiler Feed Pumps under Transient Operating Conditions" by Igor J. Karassik, George H. Bosworth and Warren D. Elston, presented at ASME Fall Meeting, October 1953, Rochester, N.Y. (Worthington Reprint RP-961).



(LOAD REDUCTION FROM 117.5MW TO 28MW)

CURVES PLOTTED FROM RAPID LOAD DROP DATA SEPTEMBER 21 1954



(LOAD REDUCTION FROM 80.0MW TO 0 MW)

Fig. 3—Above chart is a companion to Figs. 4–5

←

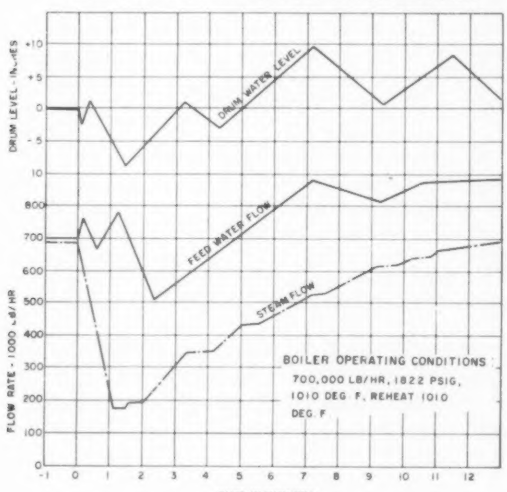
Fig. 2.—Steam flow, feedwater flow and drum level were charted by means of "souped up" chart mechanism with a 60-1 speed up gear. This makes the chart a 24 minute one and the swift changes in flow and drum level from the sudden load drop is accurately pictured

load. The exact pump capacity would then correspond to the intersection of the pump Head-Capacity curve with the system-head curve constructed with feedwater regulator valve wide open. This capacity can then be compared to the maximum safe capacity determined as on Fig. 1 and a decision can then be made regarding the relative safety of the installation. If this decision is marginal, one can choose between taking a slightly less pessimistic position regarding feedwater flow behavior and providing some protective controls which could save the situation if actual events coincide with the original assumption.

If a more optimistic approach is desired, the designer can make a study of other units in his system which have comparable feedwater controls. From this study, he can make some approximation as to what he may expect as a maximum feedwater flow value following load reduction.

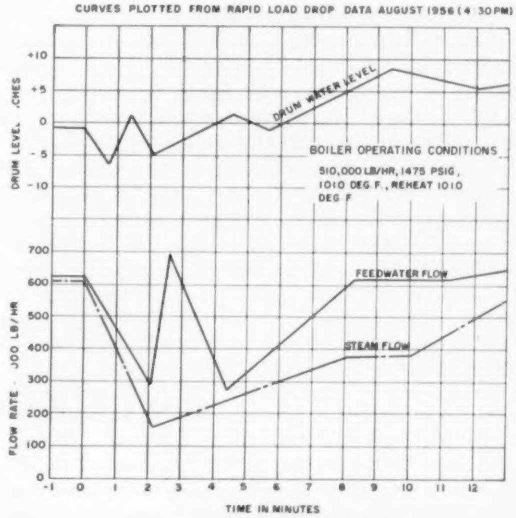
(Continued on following page)

CURVES PLOTTED FROM RAPID LOAD DROP DATA OCTOBER 19, 1955



(LOAD REDUCTION FROM 106 MW TO 12 MW)

CURVES PLOTTED FROM RAPID LOAD DROP DATA AUGUST 1956 (4:30 PM)



(LOAD REDUCTION FROM 81 MW TO 10 MW)

Fig. 4–5—In an attempt to gain more documented data such as Fig. 2 the author has sought and obtained several charts of load reduction experiences

The above are typical. The problem, however, is one of correlating the pictured data with the type and setting of feedwater controls

In your article for the Steam Power Plant Clinic—Part IX, published in the June issue of Combustion Magazine, you stated that you would be pleased to receive information on measured temperature rise as a means of controlling boiler feed pump recirculation.

Hagan has a successful installation of recirculation control based on measured temperature rise, presently installed at Unit No. 3, Will County Station, of the Commonwealth Edison Co.

R. P. Oelschlager,
Manager, Proposal Engineering
Hagan Chemicals & Controls, Inc.

The surest way to always being right is to never say anything—or at least never to state anything categorically. But this is not necessarily the best way of either learning anything new or of transmitting information that may be of value to others. The alternative is to present the latest information one has at hand and hope that any inaccuracies may be of an insignificant character and/or that you are promptly corrected by your readers.

And so it has fortunately been in the case of an answer of mine published in the June 1959 COMBUSTION in the Steam Power Plant Clinic. I had stated that I had no direct knowledge of any boiler feed pump installations embodying temperature rise controls to actuate the minimum flow recirculation. I had explained this by the fact that temperature rise controls have an appreciable time lag in registering a temperature rise such that the recirculation by-pass must be opened. This time lag could lead to dangerous operation of the boiler feed pump at extremely low flows.

After publication of this particular Steam Power Plant Clinic, I received a letter from Hagan Chemicals and Controls, Inc. pointing to a successful installation of recirculation control based on measured temperature rise in a large steam-electric central station. This installation, I am told, has been in operation for the past year and one-half.

I am advised that the potential problem of the time lag is met by the use of high-speed thermocouples and by measuring the discharge temperature in the leak-off water from the boiler feed pump balancing device. Finally, the use of a combination of electronic and pneumatic controls further provides for rapid response.

I also received a letter from the U. S. representative of the British firm of valve and control manufacturers, Hopkinsons, Ltd. indicating that temperature rise recirculation controls have been quite widely utilized in Great Britain. I note, however, in this last case that a dual set of controls is generally used in Great Britain so that the impulse given by the temperature rise is backed up by a conventional flowmeter control. Whether such double protection is justified is difficult to say.

IGOR J. KARASSIK

We have read the series of articles on the subject of anti-flash baffling with deaerators including the recent article appearing in the February issue of "Combustion" and we have the following comments:

Referring to the "Steam Power Clinic" articles, and Mr. P. H. Hardie's recent article appearing in the February issue of COMBUSTION, *these questions and answers on the subject of "Anti-Flash Baffling" in deaerators are very interesting and informative.*

It is recognized that the value of anti-flash baffling is questionable in some cases, although as reported by Mr. Hardie and others, there have been installations in the past which reported boiler feedwater pump flashing problems that were corrected by the addition of anti-flash baffling. As a matter of fact, one of the deaerator manufacturers corrected one of their installations by installing such baffling which prompted other deaerator manufacturers to incorporate this feature in their power plant plant deaerators.

From the arguments contained in the above-mentioned articles, it would appear that the use of anti-flash baffling is unnecessary and even detrimental, particularly where a Worthington Pump is being used. However, perhaps the same operating conditions will not affect other pump designs in the same manner where npsh requirements differ. Also, there are other considerations that will affect the problem, such as size of outlet piping, plant operating conditions, etc. The Japanese installation referred to in these articles is a very severe and unusual operating condition which would not normally be encountered in most deaerator installations.

I do not agree with the author that stratification of the deaerated water storage in the storage compartment can occur with various temperature gradients. Under normal operation, there is a thermal potential that would exist in the storage tank where heat would flow from the hot water to the cooler water to cause a circulation and mixing of the deaerated water storage. Therefore, the water temperature should be fairly uniform.

It does not seem wise to state unequivocally that anti-flash baffling should be completely eliminated on any and all installations. Much more work and experimental data must be done on many different conditions, before it can be said that this baffling should be, or should not be eliminated.

RALPH M. LEMEN
Manager, Heat Transfer Section
THE PERMUTIT CO.

Editor's Note:

From time to time as mail reaches us on the various Clinic subjects we shall publish and where possible also carry Mr. Karassik's reply.

American Power Conference in Review—I

The twenty second annual meeting of the American Power Conference was again held at the Sherman Hotel in Chicago, March 29-31, inclusive. Registration was excellent again with about 3000 in attendance.

Central Station Plant Design

The following were not presented at any single session but their subject matter, to our way of thinking, lent themselves to this grouping.

The "Problems in the Design of Concrete Chimneys" was presented by **David O. Thompson,** Commonwealth Edison Co., and **Max Zar,** Sargent & Lundy. During the past decade, the desire for increased pollution control, which requires taller stacks, made the concrete chimney competitive. Fortunately, by this time, methods of concrete control had greatly improved and the American Concrete Institute, after an exhaustive study, prepared a new rigid specification which became a scientific guide for all concrete chimney design in the country.

About two years ago the utility industry became aware of chimneys which were showing external streaks. These particular chimneys happened to be operating at positive pressure and this led to the belief that only pressurized chimneys have pervious linings. The change from negative to positive pressure occurs at about 75 fps exit velocity.

A research committee was set up to solve the leaky chimney problem. At the first meeting it was decided to take core samples from the shell with the stains and, just to prove that there would be no trouble, to core one of the negative pressure chimneys. Negative pressure chimney linings do not leak, the observers found, but the acid attack seems to be about $^1/_3$ as fast as for pressurized chimneys.

All the w-ll-known protective coating manufacturers were contacted in a search for a coating that would withstand temperatures up to 350 F, liquid HS_2O_4 varying from 5 to 70 per cent and the abrasion of fly ash at a velocity of at least 120 fps. Abrasion resistance becomes a factor for coatings in the gas stream.

Breechings, it was noted, which are unlined but have exterior insulation have been trouble-free for many years even with coal averaging 4.5 per cent sulfur content. Why not then install an insulated steel lining in a chimney and produce, in effect, a long breeching? This approach has been tried at Commonwealth Edison. Below we present the authors' conclusions.

All of the present type of corbel supported brick lined chimneys will probably leak gas.

Sodium silicate mortar should not be used if it is likely to reach a temperature of 50 F or less within 8 hr after installation of the brick because it flows out of the joints at the lower temperatures and weakens the lining.

Existing brick linings can be made "leak tight" by following a procedure given in detail in the paper.

With the coating materials which have been or no doubt will be developed, the authors foresee several methods of constructing chimneys: (1) A redesigned concrete shell coated but with no insulation or lining (limited to the few short, large diameter installations). (2) Steel lining with attached insulation. No coating on concrete shell but the exposed surface of liner may be coated if operating experience indicates necessity. (3) Redesigned corbel to eliminate troublesome corbel blocks. Encapsulated insulation with brick shoved against the tacky final coating on insulation. (4) Independent brick lining.

J. L. Menson, W. L. Harding and **E. P. Petit,** Combustion Engineering, Inc., chose as their paper "Predicting Tube Life in High Temperature Boiler Installations." In high temperature installations where tube wastage develops this situation is not basically a Boiler Code problem for several reasons. The first is that the Boiler Code is essentially a safety code and in modern boiler design, the rupture of a superheater or a reheater tube seldom constitutes a safety hazard to personnel. The second and more basic reason is that the solution to the problem does not lie in any modification of either the Code design rules or the allowable stresses. The circumstances of tube wastage are extremely variable. The owner of a boiler in which some tube wastage is occurring is, therefore, confronted with a problem for which the Code rules provide essentially no guidance.

This paper describes methods to provide a reasonable basis for predicting the life before rupture of tubes in which wall reduction by wastage has been measured:

The first step is a review of all available data on rupture life vs. temperature and stress for the particular tube material involved. By trial-and-error selection of the C term in the Larson-Miller parameter, it has been found practical to correlate this rupture data as a straight line.

The second step, the evaluation of wastage occurring in a particular boiler, is most readily established by careful measurements of the outside diameters of the tubes and comparison with the initial tube size as installed.

The third step is to calculate the stress in the tube wall. This is done by the same ASME Code tube formula used in the initial boiler design.

The final concept needed to tie together this evaluation of material rupture strength and the increasing applied stress is the "fraction of rupture life" hypothesis. This states that spending a time interval Δt at some stress σ and a constant temperature will consume a fraction E of the total life-to-rupture t_r, such that $E = \Delta t/t_r$. If the material is subjected to a number of stresses (σ_1, σ_2, . . . for various time intervals (Δt_1, Δt_2, . . .), rupture will occur when the sum of the various fractions of life reaches unity.

J. C. Beres and **R. W. Potts,** Commonwealth Associates Inc., teamed up to present the paper, "Selection of Boiler Feed Pumps and Drives." In the paper the authors give the results of a study that was made in selecting the most economical boiler feed pump and drive arrangement for a 265,000 kw unit, one of two to be installed at a location in the Great Lakes region. Steam conditions for these units are 240,000 psig, 1050 F at the throttle and 1000 F reheat. Each unit will be a 3600 rpm cross-compound four-flow type. The first unit is scheduled to be placed in operation in 1962 and the second in 1963.

As a basis for evaluation, the following criteria were established:

(1) The net kilowatt capability with any of the boiler feed pump drive arrangements to be the same.

(2) An average unit capacity factor of 0.63 for 35 year life.

(3) Expected coal cost of 40 cents per million Btu at the startup date and increasing at the rate of 2 per cent per year for the 35 year life.

(4) Comparable reliability of arrangements as far as possible. When this is not possible, an outage penalty for the anticipated difference in down time is applied.

(5) Main turbine-generator exciter to be driven off the main turbine-generator shaft with any boiler feed pump drive arrangements.

(6) Fixed charges to be at a rate of 15 per cent.

Since various combinations of boiler feed pumping equipment have been proved to have comparable reliability, a study was first made to determine the most economical combination with each of the three basic drives. These in turn were compared to arrive at a final selection.

The most economical motor drive arrangement would be with two one-half size, 5400 rpm constant speed pumps, each driven by an 1800 rpm motor through a speed increaser.

The arrangement with two one-half size auxiliary turbine-driven boiler feed pumps was selected as the auxiliary steam turbine drive arrangement, and the most flexible and economical turbine-generator shaft drive arrangement was held to be with two one-half size, 3600 rpm pumps each driven through its own fluid coupling.

These three selected boiler feed pump arrangements, each with a different method of drive, were next compared. A table showed the variations in the basic specifications for the turbine-generator and for the associated pumping equipment on the basis that the net kilowatt capability of the turbine-generator would be the same with all three boiler feed pump drive arrangements. Keeping the net capabilities the same required a minimum of major power cycle equipment changes and, thus, comparison of the boiler feed pump drive arrangements was greatly facilitated.

A second table showed the equipment costs, excess cost differences and penalty cost differences of the selected drive arrangements. It is evident from the table that the main turbine-generator shaft drive arrangement is obtainable at a cost reduction of $229,500, compared to the motor-driven arrangement and at a cost reduction of $178,000, compared to the auxiliary turbine drive arrangement. The shaft driven variable speed pump arrangements, unlike the motor-driven variable speed pump arrangements, have no generator, transformer and motor losses to add to the power requirement.

Ernest A. Sticha, Edward Valves, Inc., covered "Structural Stability of Commercial Wrought Austenitic Steels for Power Plant Piping to 1450 F." Austenitic steels in high-temperature power piping systems lose impact resistance because of structural changes taking place during service. Data previously available did not reveal the ultimate extent to which properties deteriorate. About eight years ago, the Edward Research and Experimental Laboratories started a program to fill this data gap with meaningful tests under carefully controlled conditions.

The author's summary stated:

(1) Commercial grades of austenitic steels may lose considerable room-temperature impact resistance during long-time, high-temperature exposure.

(2) Loss of impact properties is usually greatest at 1450 F (the highest temperature of testing) but this varies with composition, etc. Alloys such as Type 321 and 316F show greatest loss in 1250 to 1350 F range.

(3) Deterioration of properties in the early stages of exposure is due to carbide precipitation and is relatively minor. The greater losses produced by longer aging are the result of sigma formation.

(4) The extent to which impact properties degenerate seems to be a function of composition, with alloys of lower chromium equivalent showing less loss.

(5) Impact characteristics of aged samples vary with test temperature, the values being lower at subnormal temperatures and higher above ambient.

(6) Steels less susceptible to embrittlement by elevated temperature exposure may be obtained by lower equivalent chromium content, either directly or indirectly by higher carbon and/or nitrogen in the steel.

(7) Phase diagrams showing the limiting chromium content for sigma formation to be about 18 per cent may have to be revised to a lower limit.

(8) A higher molybdenum austenitic steel, Type 317, suffered catastrophic oxidation at 1350 and 1450 F after about 46,000 hr.

(9) Surface attack, probably nitriding, was noted in some of the steels aged at 1450 F but none was observed at 1350 F after 49,000 hr exposure.

Wm. E. Wendover of American-Standard Industrial Division, spoke on future applications of "Mechanical Draft Fans." Speaking of the trend of fan requirements, Mr. Wendover said that since 1950, pressurized firing has been increasingly favored so that today the majority of boilers use this method. Three or four forced draft fans may be required on many larger units.

Most boilers of 1970 will probably be pressurized. Many will require four to six forced draft fans due to their large size.

The addition of heat recovery equipment and the use of higher velocities have caused a continual increase in fan pressure requirements. It is interesting to note that average fan power requirements per pound of steam have increased significantly since 1940, even though the types of fans now used are 10 to 15 per cent higher in efficiency.

The author thought that these important areas of application need study: (1) optimum number of fans per unit, (2) method of capacity control, (3) fan design limitations, (4) fan noise, (5) equipment and ductwork layout.

Covering the number of fans per unit recent experience indicates that boilers generating four to five million pounds of steam per hour can be handled adequately with four fans of types commercially available. This would indicate that boilers of up to $1^1/_4$ million lb of steam per hr capacity can be handled with one fan, or those of $2^1/_2$ million lb of steam per hr with two fans. With today's improved fan designs and fabrication methods, reliability of the draft system when properly instrumented and maintained, can easily be as good as that of the other major components of a steam generating unit.

On fan control methods the speaker thought that, historically, fan control methods have been evaluated primarily by comparing first cost of fan and control against power savings at reduced loads. Present and future fan requirements however, call for a more erudite treatment of control selection, with emphasis on some less tangible features of control. One factor which should be investigated is the comparative cost and availability of drives to actuate the fan control. Recently there have been cases where the torque requirements of dampers or inlet vane controls on large high-pressure fans could not be met by commercially available standard control drives.

The noise level of today's high pressure fans is becoming more important. The method of control used has a marked effect on the noise level at reduced loads. This will be discussed in detail in a later section.

The inertia of the fan rotor may determine the choice between constant speed (damper or inlet vane) and variable speed control. As boiler capacities and fan requirements increase, the fans selected for best efficiency also increase in size. Maximum fan size is limited by the size of assembled wheel which can be fabricated and shipped by practical methods. Fan housings can always be split into convenient pieces for shipping, but it is unlikely that highly stressed rotors will ever be shipped in parts for assembly in the field.

During the next decade, we can expect to see shipped a number of fan wheels in diameters of 10 to 12 ft, with complete rotor weights up to 35 tons. Careful planning will be necessary to provide correct erection procedures and equipment for such fans.

The control of noise from mechanical draft fans is now and will increasingly become one of the biggest headaches in the field of application. The present state of the art is about where the dust collection art was thirty years ago. The problem is similar, for both are side effects and nuisances which cost money to correct, yet provide little return on the investment.

Peaking Studies

The currently heavy interest in the application of peaking units to established systems was given a full session treatment.

Lester B. LeVesconte and **Tor Kolflat,** Sargent & Lundy, approached the problem with the paper "How Shall We Meet Peaking Requirements?" To correctly evaluate comparative costs and other merit factors of different types of peaking capacity, it is essential to determine the magnitude of such extra peaking to normal load and unit size, the expected number of annual operating hours, the daily and seasonal duration of such peak loads. There are various design and manufacturing margins incorporated in major power plant equipment to insure equipment meeting the specified capacity and performance. Sometimes additional capacity found is available by virtue of these margins. The different failures which actually occur in power plant equipment illustrate, the authors pointed out, that, in spite of all calculations, there is still uncertainty or ignorance and the owner must accept the risk.

Another possible method of obtaining peak capacity is to purchase interim capacity from neighboring utilities. Based on average coal steam plant costs short time peak capacity might be purchased for about $20 per kw a year demand charge. In all cases, though, the cost of increment capacity is about 20 to 30 per cent less than the average cost of a conventional steam power plant and the authors suggest a review of the comparative merit of peak capacity as an increase in size of contemplated units instead of separate peak units. However, where a large amount of peak capacity is required only for a few months, and could be completely shutdown for eight to nine months, say, low cost, low efficiency units, sacrificing stack temperature, and feedwater heating, vacuum, etc., should prove attractive. The authors cited a study made for a 100,000 kw gas or oil fired unit which indicated approximately 20 per cent saving in capital cost compared with an efficient unit of the same size. For new steam plants which are in the design stage, the authors believed it perfectly possible to obtain 10 to 20 per cent short time peak capacity at a cost well below both average normal and incremental capacity cost.

Messrs. Kolflat and LesVesconte then outlined certain of the possibilities and commented briefly on them. In the main these methods were: (1) shutting off of feedwater heaters, (2) steam pressure increase, (3) bypass of turbine high-pressure end, and (4) change in fuel during peak hours. Following these possibilities which exist within the plant the authors mentioned in passing specialized peaking capacity methods such as pumped storage, gas turbines and the internal combustion engine.

The next paper "Low-Cost Incremental Peaking Capacity" by **F. A. Ritchings** and **R. R. Bennett,** Ebasco Services Inc., opened with the comment that in the past,

each new generating unit a utility company installed has been generally at least as efficient as its immediate predecessor. This philosophy has been justified economically by the availability of significant heat rate improvement at reasonable cost even though the more efficient new unit, by being base loaded initially, resulted in displacing each of its predecessors upward in the load-duration curve and reduced the use and average loading of earlier units.

The industry is now at a plateau where further substantial reductions in heat rate can be accomplished only at a significantly greater cost and, at present, with perhaps a lesser degree of reliability than experienced with currently installed units. Thus we must re-examine past practices. Over the past two decades heat rate reduction has averaged about 3 per cent per year. The annual reduction in heat rate for the foreseeable future is estimated to average only about 1 per cent per year.

As a result the economic solution for systems that already have a high proportion of their production facilities in modern, efficient reheat units is to schedule installation of new units so that the present reheat units are held in the lower portion of the load-duration curve for a longer period of time. Since a peak presents only a few hours of operation at high system loads, the fixed charges on production plant investment for energy production in this peak are many times more significant than the cost of operating labor, maintenance and fuel. The conditions the authors proceeded to describe are for a utility system having a 63 per cent average annual load factor. Many systems have a much lower load factor and it is obvious that the lower the system load factor the more pronounced these cost conditions become.

There are several means by which capacity can be added to a system to meet a particular load service. They include (1) high-efficiency base-load unit, (2) low-cost and relatively low-efficiency nonreheat unit, (3) incremental-capacity units, (4) peak-load units such as diesel, pumped-hydro storage or gas turbine. These various methods were then illustrated by special cases. The authors provided a table which summarized the estimated investment requirements for each of the different schemes considered. The 250-Mw unit with or without peaking capability is assumed to be a new plant on a new site. Prices for the turbine generator and steam generator are the averages of those obtained from the manufacturers while the costs were estimated for the other equipment and material affected.

From this tabulation it may be noted that, at present price levels, peaking capacity may possibly be obtained at an incremental cost of about \$30 to \$39 per kw for gas firing depending on the scheme selected.

This compares to the estimated cost of \$102 per kw for the base plant. For comparative purposes, a 100-Mw gas-fired unit installed specifically for peaking purposes is estimated to cost \$90 per kw. This cost would increase to about \$93 per kw for an oil-fired plant.

The cost of a high efficiency reheat, gas-fired plant of 325- to 350-Mw capacity instead of the 250-Mw unit considered was also estimated and the incremental cost over the base 250-Mw plant is approximately \$75 per kw.

In a coal-fired system where coal is also used for peaking it is necessary to add pulverized equipment and extend the coal handling equipment as substantial peaking capability is added to the base 250-Mw unit. The cost for this plant is estimated to be \$36 to \$41 per kw. The cost of the base coal-fired plant is estimated to be \$122 per kw.

In a coal-fired system where oil is used for peaking, the incremental cost for obtaining peaking capacity is estimated to be \$33 to \$48 per kw except for Scheme G. In this scheme no additional pulverizing capacity is required to obtain the peak capability with coal firing due to the margins built into the base plant. However, when the peak is obtained with oil firing then additional equipment must be added. The incremental cost of Scheme G with oil firing is estimated to \$41 per kw as compared to \$39.50 for coal firing at the peak.

Since these costs are based on estimated prices for steam generator and turbine generator and recognizing that each situation will be accompanied by its own special circumstances, it is considered that the incremental cost of peaking capacity may be somewhat higher than indicated.

The main disadvantage of these schemes is the decrease in heat rate at the normal turbine capability and also at lower loads. If the loss in heat rate is 1 per cent, then a capitalized cost of \$250,000 is chargeable to these systems assuming 63 per cent load factor, \$0.25 per million Btu fuel cost and 14 per cent fixed charges. This would raise the incremental cost per kw by \$2.50 to \$4 depending on the scheme selected but would still result in low incremental cost for the peaking capacity.

J. O. Stephens and **B. L. Lloyd,** Westinghouse Electric Corp., collaborated on the paper, "The Economies of a 22,000 Kw Peaking Gas Turbine." The authors present in some detail the results of a specific analysis which was made on a system to answer this question: "Should the next unit be a gas turbine or should system expansion be continued by conventional base load steam units?" The paper then went on to describe characteristics of the 22,000 kw gas turbine plant included in the study.

The procedure adopted for this evaluation was first to define the characteristics of the model system. Present loads, installed generation, and production costs were defined. Future load growth trends and a criterion for installed reserve were established.

A base expansion pattern was developed for the next 12 years with new generation requirements being met by a sequence of base load steam units. A peaking pattern was developed with the first unit being a block of gas turbine capacity. This was followed by the same sequence of conventional units as in the base plan.

Capital requirements and annual production costs for each of the 12 years were established for both the base and peaking plans. Comparison of costs for the two plans permitted evaluating whether the first block of gas turbine capacity could be economically incorporated.

Editor's Note:

The major share of the papers abstracted at the Power Conference will be published in the next month s issue.

LeRoy F. Deming, U. S. Navy Dept., Bureau of Yards and Docks, reported on "The Factory Fabricated Coal

Fired Boiler." Mr. Deming prefaced his remarks by saying that the Navy Department's Bureau of Yards and Docks experience in the erection of field erected boilers indicated the following conditions:

(1) The coordination of the delivery of principal items of equipment with the construction program, particularly when equipment was Government furnished, was generally not too successful.

(2) Field erection was costly because of the multiplicity of trades involved.

(3) Scheduling of erection work was difficult, productivity of field labor was low and subject to interruption by inclement weather, and it was difficult to maintain a good quality of workmanship.

(4) The above factors all contributed to inefficient plants and costly maintenance, particularly in the lower capacity jobs in remote locations. Unsavory results were reflected by increased unit costs and lowered integrity of the service the plant was intended to render. The record indicates that each of the above factors served as stepping stones to higher costs to the Government for the completed plant.

About 1951 the Bureau of Yards and Docks adopted a standard specification providing for factory fabrication of oil and gas fired water tube boilers in capacities from 10,000 to 27,000 lb per hour.

As a result certain specific benefits became available:

(1) The cost of the boiler delivered and erected on the owner's foundation was greatly reduced, in some instances as much as 50 per cent. Shipping time was also greatly reduced.

(2) Automatic controls of the types previously limited to boilers burning only natural gas or distillates were quickly developed together with necessary safety devices to provide fully automatic operation on residual fuel oil.

(3) Heat release rates formerly considered as maximum were found to be too conservative. Satisfactory operation at much higher rates of heat release were found possible in the long narrow furnace dictated by shipping limitations. The specified maximum heat release has been modified upwards, in some cases as much as 75 per cent, and the maximum capacity available for railroad shipment has somewhat more than doubled for boilers conforming to the current military specification.

While factory fabrication was adapted for oil and gas fired boilers, with the very favorable results enumerated above, the situation changed only slightly for the coal fired boiler in the same general capacity range. The unsavory conditions enumerated in the early part of the paper still prevailed and until about 1954 the extension of automation to the coal fired boiler did not appear possible due to the excessive cost of adapting mechanical ash scavenging to the coal fired job.

The Bureau purchased shop fabricated boilers equipped with newly available mechanical ash discharge spreader stokers and instituted a test and development program. Boilers of the longitudinal and transverse drum types were tested.

Mr. Deming summarized, as follows, the conclusions drawn from the test and development study:

(1) The development program initiated by the Navy Department, Bureau of Yards and Docks, has demonstrated the feasibility of factory fabricated boilers equipped for spreader stoker firing of coal and readily adaptable to oil as an alternate fuel in capacities from 10,000 lb per hr to approximately 45,000 lb per hr.

(2) Boiler performance in each instance exceeded design expectations and indicates greater furnace absorption rates are identified with the long horizontal furnace. Successful operation on a wide variety of coals also exceeded design expectations and indicates that factory fabrication does not in any way limit performance within capacities to which it is adaptable. The lower temperature of gases leaving the boiler indicates that increased limits on heat liberation are possible. This condition is further confirmed by reports on two 35,000 lb per hr longitudinal drum D type factory fabricated boilers installed by another manufacturer in Ohio. These were also equipped with mechanical ash discharge spreader stokers, but grates are of the reciprocating self-cleaning type and the stoker was assembled in the field.

(*It should be noted, Mr. Deming pointed out, that the Bureau is considering discontinuance of heat release in terms of Btu per cubic foot of furnace volume as a criterion of design and substituting therefore a specified maximum rate of furnace heat release per square foot of radiant heat absorbing surface.*)

(3) Factory fabricated coal fired boilers equipped with spreader stokers have been quoted to prospective purchasers within the last year for $3.33 per pound of generating capacity. This is for the boiler and stoker complete with all fittings, accessories and automatic combustion control, in the general capacity range of 50,000 lb per hr, delivered in the middle west.

(4) Mechanical ash discharge spreader stokers factory fabricated can be purchased for approximately 25 per cent above the cost of dumping grate spreader stokers or of the side dump single retort stokers. This makes possible full automation in the lower capacity coal fired boilers, 10,000 lb per hr and upward.

(5) Finally, the mechanical problem of fitting a factory boiler and stoker together in the field has been found to be so encumbered with difficulties of clearance, fit, and adjustment that it is recommended no one should normally expect satisfactory results unless at least the first units of any design are fully fitted together in the factory before shipment to the field.

Messrs. **H. M. Reyner,** Western Electric Co., and **L. P. Copian,** Riley Stoker Corp., presented a paper on "Slag Tap Boiler Performance Associated with Power Plant Flyash Disposal" at the Fuels Session on Wednesday afternoon. The authors described how the problem of flyash disposal from pulverized coal-fired boilers is becoming increasingly difficult and more expensive. Many plants are sluicing their flyash into sloughs or waste areas and these areas are fast disappearing. For other plants the usual disposal areas, such as local quarries, are refusing to accept flyash in the waste disposal area as the result of public sentiment.

This problem became increasingly acute in 1955 and 1956 at the Western Electric Co., Hawthorne Station.

The authors noted that early in the design stage it became apparent that the difference in the cost between flyash disposal and slag removal was large enough to make a boiler with slag tap design attractive. The additional investment in auxiliaries for slag disposal was justifiable, and there was an additional advantage in reducing the nuisance associated with flyash disposal.

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This paper discusses some of the more important and interesting aspects in the use of a digital computer in a study to determine the economic signs of a condenser. The conventional formulae used by condenser engineers have been placed in the computer to obtain the total operating and investment cost for the 35-year life of the condenser. The condenser with the lowest summation of costs is then selected. This article has been divided into two parts. The first, presented now, deals almost exclusively with the condenser selection problem. The second part, to be published next month, gives the actual outline of data and formula for the condenser evaluation.

Economic Sizing of Condensers Through the Use of the Digital Computer*

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TO PROPERLY size a condenser, consideration must be given to using many variables such as tube lengths and diameters, heat transfer surfaces, and water velocities, as well as turbine loads and circulating water temperatures. To calculate just a few condenser sizes requires a very large amount of labor and a certain amount of short cutting to bring the work within practical bounds. Even though the work is conscientiously done, the size selection is often under question.

It has been felt that for the large units which we are now installing, the longhand evaluation is no longer satisfactory. We realized that for a large (300 Mw) unit even a small improvement in condenser vacuum would result in a considerable savings in operating cost during the 35-yr life span of the unit, and it was decided that a more rigorous and refined evaluation should be employed in determining the optimal condenser size. It was for this reason that the Commonwealth Edison Company made use of a digital computer in the selection of several of their recent condensers.

The use of a digital computer still involves a considerable amount of time and labor, but it does increase the accuracy and permits the study to cover a great many more sizes and variables.

Computer Selection

There are a number of digital computers available throughout the country. The IBM 650 is probably the most widely used computer for engineering problems. This was the one first selected by our computer specialists, but it proved too small for this application.

The problem was reprogrammed and then solved by the IBM 705 computer. Due to the large number of variations it was necessary to solve for 2860 separate condensers. Besides requiring a considerable amount of time by the computer specialists it took about 30 machine-hr. At $200 per hr, this came close to $6000 and this amount is difficult to justify.

To reduce this cost on the next similarly sized unit, those variables which were considered to lie outside the range of the problem were eliminated. The number of condensers solved was reduced to 600, or roughly one fourth the number in the first study, and this resulted in a comparable saving in computer hours. To reduce the cost still further on future studies, the 704 should probably be considered. It is understood that the 704 should solve a similar problem in less than an hour.

For those who are not familiar with digital computers

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some explanation is in order. The 705 has been designed for accounting or business purposes, with a large input and output, but a small computation section. The 650 and 704 are primarily intended for engineering problems. Both of these computers have small input and output sections with a large computation section, but the 704 can take a much larger problem than the 650 and, therefore, this type of computer would be ideal for solving condenser problems.

Economics of a Condenser

The function of a condenser is to produce an economical back pressure on the turbine exhaust. The back pressure is dependent upon a number of variables, the more important ones being condenser surface, circulating water velocity and temperature, and steam flow to the condenser. Since the heat rate of the steam cycle and the capability of the unit is very sensitive to the condenser back pressure, an economic value can be put on the heat rate and capability in terms of fuel cost and plant investment, respectively. Therefore, it is the proper combination of all these factors which determines the optimum condenser size.

The following remarks discuss only those items which are unusual, or treated in a manner other than normal.

Cost of Coal

Since there was some doubt that one could peer into the future and forecast the cost of fuel with accuracy for the life of the equipment, until recently all calculations for new equipment were based on present fuel costs.

By charting the past 50-yr coal cost history, it became possible to develop a projected fuel price. Fig. 1 shows this chart, the trend line, and the projected coal cost. The trend line in solid black has been determined by use of the standard trend formula which is by the "sum of the least squares method."

Since the transportation of fuel is such a large proportion of the total fuel cost, it was necessary to adjust the average cost according to the station location. The curve for the unit in the study starts at the actual present day station fuel cost and gradually assumes a position parallel to the trend line.

Recent units have been evaluated by using the projected fuel cost.

Fig. 1—Charting coal costs for the past fifty years gave the authors a trend line against which they could develop future coal costs

Capability Evaluation

Economic studies show that large exhaust annuli are justified for the circulating water temperatures and load factors experienced on most of the Edison units that have recently been purchased. Units with large exhaust annuli show an appreciable increase in capability with a decrease in back pressure.

A table showing the capabilities of the turbine with wide open throttle at various back pressures is used to evaluate the capability gains at full plant cost. Some may question the advisability of using the full plant cost and may prefer to use some adjusted or incremental cost. It is well that this item be given serious consideration, since the condenser size is quite sensitive to the evaluated capacity gain. In this study capability gains have been calculated at the summer circulating water temperature.

Circulating Water Temperature

The nine-year average circulating water temperature (1950 to 1958, inclusive) for each month of the year was obtained from the station records.

Past practice has been to average the monthly temperatures and then to use the yearly average. It was felt, however, that this might not give an accurate determination of the total fuel consumed by the unit. Since some of the monthly averages were very close, those that were close were averaged and grouped:

December, January, February and March	38.87 F
April and November	45.62 F
May, June and October	54.50 F
July, August and September	65.92 F

Heat Rejected

In sizing condensers, engineers normally use 950 Btu per pound of steam as the heat rejected to the cooling water. It is questionable whether this figure is representative of the actual heat rejected for the loads and back pressures at which the unit would operate. In this study the heat rejected was obtained from the turbine engineers for two loads and five turbine exhaust pressures. By plotting heat rejected versus load back pressure as a parameter, the heat rejected at all loads was obtained. Using these curves, a new set of curves were plotted showing heat rejected versus back pressure with the economic load points as a parameter. From this final set of curves, the heat rejected was obtained at each of the selected load points for all back pressures. These figures were then used in calculations rather than 950 F.

Condenser Flow

To determine the condenser flow, the throttle flow is obtained at some given back pressure (usually 1- or 1.5-in. Hg) and at one or more load points, and these should be stated in the turbine proposal. In this study, the throttle flow at rated load and back pressures between 0.3- and 3.5-in. Hg was determined from the equation:

$$\text{Throttle flow} = \text{nominal rating of turbine} \times \text{throttle steam rate} \times \text{correction factors}$$

The correction factors (Reference 2 at the close of the article, Section 7.2, Figs. 7.1a to 7.1f) are for the throttle temperature, exhaust annulus area, back pressure, rated

generator output, temperature of feedwater leaving the heater at the reheat point, and feedwater temperature rise on the heater above the reheat point.

The condenser flow at all loads and back pressures can be determined from the above throttle flows by using the following equation:

Condenser flow = throttle flow at rated load × per cent of turbine load × correction factors

The correction factors (Reference 2, Section 7.2, Figs. 7.3a and 7.3b) are for the final feedwater temperature at rated generator output and exhaust pressure, and per cent of rated generator output. This information was then tabulated and put in the computer program.

Load Duration or Capacity Factor

The load duration or capacity factor of the unit is usually the subject of much discussion. For this study the annual capacity factors for the system were reviewed, and the average of these factors was adjusted to reflect the effect that the cost of coal and the operation of the "automatic dispatch system" (ADS) might have on the loading of the unit under study. The ADS allocates the load to the various units on the system in accordance with the relative cost of the output, and this involves unit efficiency, fuel cost, line losses, etc.

For this study it was the judgment of the engineers that the capacity factor over the 35-yr life of the unit would be 48 per cent.

Using this capacity factor, the loading of the unit for the four loads and three periods selected would be:

Load		1st Period of 10 Yr,	2nd Period of 10 Yr,	3rd Period of 15 Yr,
Mw	Per cent	Hr/Yr	Hr/Yr	Hr/Yr
325	100	700	440	0
284	87.5	3330	2150	790
203	62.4	2890	2670	1840
144	44.5	970	1490	2450

Turbine Heat Rate

Fig. 2 shows the turbine heat rate plotted against vacuum with load as a parameter. The heat rates, as read from these four curves, were tabulated at intervals of 0.1-in. Hg and then placed in the computer program. The curves for the high loads are quite flat and they show little gain at the lower back pressures. The curve at rated load (325 Mw) shows a slight increase in heat rate as the pressure drops.

For units of similar characteristics, it has been the Operating Department's practice to reduce the amount of circulating water for such conditions, thereby improving the heat rate and saving the pumping costs. Trial calculations indicated that one-pump operation was economical at loads of 325 and 284 Mw for the average circulating water temperatures of 38.87 and 45.62 F, respectively.

Computer Print-Out

The application of the digital computer to this problem makes it possible to obtain a large quantity of results which would not be available if longhand and slide rule methods were used. The real problem in reference to print-out is to determine in advance the results desired, and then to set the program to obtain them.

Actually, the problem is being programmed to determine the condenser size which will produce the lowest overall cost. To accomplish this, it is only necessary to bring out summation of all costs and a combination of figures that identify condenser size and proportions.

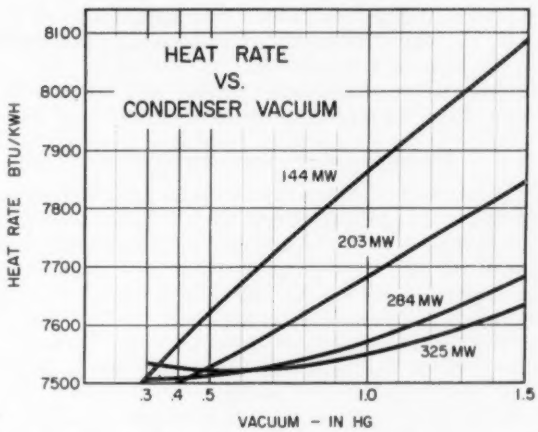


Fig. 2—Turbine heat rates for various loads and back pressures were plotted to be fed into the computer

On the first condenser problems, we brought out a rather meager amount of information. Therefore it was difficult to analyze results, note trends, and do a good job of answering many questions our colleagues raised.

The print-out on the latest study gives the engineer considerably more information as follows:

Identifying Information

1. Condenser surface.
2. Cooling water velocity.
3. Tube length.
4. Tube diameter.

Calculate total cost:

(a) Two-pump operation, capability at 100 per cent of plant cost.
(b) Combination one- and two-pump operation, capability at 100 per cent of plant cost.
(c) Two-pump operation, capability at 90 per cent of plant cost.
(d) Combination one- and two-pump operation capability at 90 per cent of plant cost.

Additional Information

1. Condenser cost (shell plus tube cost).
2. Cost of tubes.
3. Weight of tubes.
4. Friction head of condenser and tunnel.
5. Pumping horsepower (two pumps).
6. Gallons per minute used, two-pump operation.
7. Gallons per minute used, one-pump operation.
8. Total investment cost (present valued).
9. Total pumping cost for three periods (present valued).
10. Fuel cost for each of three periods (present valued). Two pump operation and combination one- and two-pump operation.
11. Condenser vacuums for all load points, cooling water temperatures for two-pump and combination one- and two-pump operation.
12. Flow of 70 F cooling water.

13. Capability (Mw) with 70 F cooling water.
14. Present value capability evaluation at 90 and 100 per cent of plant cost.
15. Total cost for each pumping arrangement excluding capability (present valued).

Discussion of the Computer Program

The problem was programmed for an IBM 705 digital computer, Model II. The first part of the program uses approximately 20,000 characters of high speed core storage and one magnetic tape for output. In addition, two magnetic tapes are used for check-point information so that the program may be interrupted at any time or may be restarted in case of machine failure. The program's second part uses one input tape, the output of the first part, and one output tape.

Due to the variations in the decimal point, fixed point scaling was impractical. The calculations, therefore, are made in the floating decimal point mode in which the computer itself keeps track of the decimal point. Each floating point number consists of two parts: a mantissa, a number between 0.1 and 1, and an exponent which is the power of 10 that restores the mantissa to its true fixed point form by multiplication. In the computer representation the number 50 is added to the exponent in order that the exponent will always be positive.

Examples:

Fixed Point	Floating Point Form	Computer Form
-12.345678	-0.12345678×10^2	$521234567\overset{-}{8}$
0.00012345678	$0.12345678 \times 10^{-3}$	$471234567\overset{+}{8}$

The first two digits are the exponent; the last eight digits are the mantissa. The sign of the number is superimposed over the least significant digit.

The calculations were divided into two parts and the program split into two subprograms. The first computes the total cost without capability evaluation; the second computes total cost including capability credit.

The computer flow diagram illustrates the logical order of the program steps. The numbers to the right on Fig. 3 refer to the numbered descriptions below:

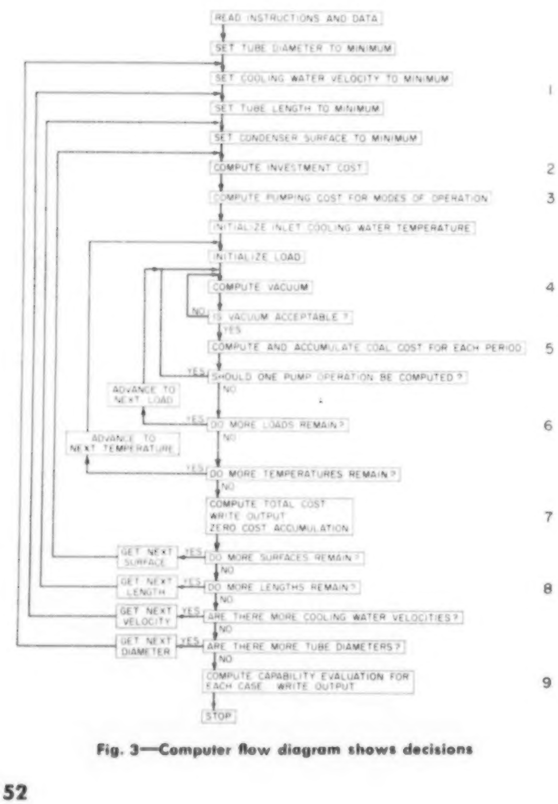


Fig. 3—Computer flow diagram shows decisions

1. The program begins with the condenser having the minimum values of the four basic variables, tube diameter, cooling water velocity, tube length, and condenser surface.

2. First, the investment cost is calculated by summing up the computed costs in the following order; namely, (1) shell, (2) erection, (3) erection supervision, and (4) the cost of tubes, (5) circulating water pumps, (6) circulating water pump motors, and (7) pipe, valves, fittings, and screens. The cost of the circulating pumps involves water flow (gpm), and tunnel and condenser head loss.

3. Next, pumping cost is calculated for each of the three periods. Each period is divided into two separate parts, the first covers two-pump operation at all times and the second covers one-pump operation at the two highest loads with the two lowest circulating water temperatures as well as two-pump operation at all other loads and temperatures.

4. The most complicated and time consuming section of the program is the calculation of the condenser vacuum and the coal cost. Starting with the minimum circulating water temperature and the highest load, the program finds the corrected heat transfer coefficient Computation of the vacuum is done by an iteration process. Assuming an arbitrary initial vacuum of 1.0-in. Hg, the program computes a new value for the vacuum. If this new value is not within 0.01-in. Hg of the old value, the new value is used to compute another vacuum and the process is continued until the final vacuum satisfies the test. Normally, about three iterations are necessary. The heat rejected per pound, the total steam flow to the condenser, and the vacuum for the steam temperature are all stored in table form. Linear interpolation between tabular values was held sufficiently.

5. After the vacuum is found, the turbine heat rate is computed from the table and the appropriate factors of boiler efficiency, hours, etc., are selected and the coal cost calculated for each of the three periods.

6. The vacuum and the coal cost calculation is now computed for each temperature and load combination and for each pump operation; the costs are then accumulated for each period and each operation mode.

7. Lastly, the costs are summed to get the total cost without capability. At this point the results of the problem are written.

8. The computer will now advance to the next larger condenser surface area and perform a similar calculation. After the largest condenser surface area is reached, the surface is reset to its minimum value and the tube length is increased. This process is continued until each combination of the various condenser surface areas, tube lengths, cooling water velocities, and tube diameters are considered.

9. In the second part of the program, the vacuum for the maximum load condition is computed at 70 F water

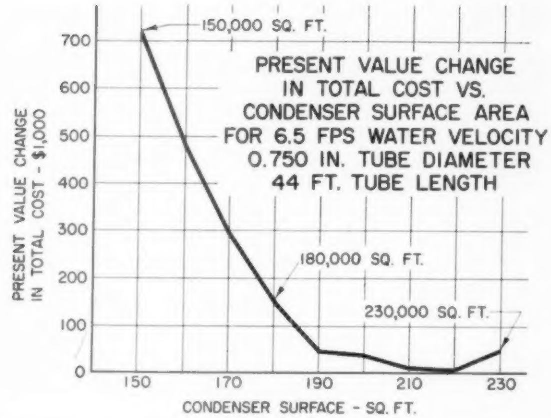


Fig. 4—The above plot showing the effect of enlarging the condenser surface with all other factors constant permits equating surface area against capability and fuel

temperature, and by using the results of the first section of the program the total cost is adjusted by the capability credit.

Study Results

The final results of the calculations performed on 650 separate condensers are shown in Figs. 4–6.

Fig. 4 shows the change in total cost as the condenser surface area is varied while the tube length (44 ft), water velocity (6.5 fps), and tube diameter ($^3/_4$ in.) are held constant. This curve shows that the investment for additional surface area can be made on the basis of fuel and capability savings as the area is increased from 150,000 to 220,000 sq ft. For condensers larger than 220,000 sq ft the total cost increases; therefore, the investment cannot be justified. A curve of this shape is obtained for each tube diameter, tube length, and water velocity.

Fig. 5 shows the change in total cost in a series of loops as the surface area, tube length, and water velocity are varied, and the tube diameter is held constant at $^3/_4$ in. In general, the trend is for a decrease in total cost as the water velocity is increased from 6.5 fps to 7.5 fps. Operating experience will determine the possible range of the water velocities. At 6.5 fps there is a sharper increase in cost for longer tube lengths than at 7.5 fps. Loop amplitude decreases for higher velocities.

Fig. 6 shows the change in total cost in a series of loops as the surface area, tube length, and the tube diameter are varied when the water velocity is held constant at 7 fps. In general, the trend is for a decrease in total cost as the tube diameter is increased from $^3/_4$ to 1 in. The $^3/_4$-in. tube diameter shows a sharper increase in cost for longer tube lengths than with the 1-in. tube. The amplitude loop decreases for larger diameters.

The study shows that a 180,000 sq ft condenser with a 40-ft tube length, 1-in. tube diameter, and a 7.5-fps water velocity would be the most economical. The 7.5-fps water velocity was considered to be in the range of possible tube errosion and was, therefore, eliminated. The next most favorable condenser with a lower velocity (7.0 fps) was the one having a surface area of 180,000 sq ft, a 40-ft tube length, and a 1-in. tube diameter. The 40-ft tube length resulted in a space limitation.

Upon careful consideration of all these limitations, the decision was made to purchase a 180,000 sq ft condenser having a 44-ft tube length, a 7-fps water velocity, and a 1-in. tube diameter.

Desirable Improvements for the Study

1. To develop the tables covering heat rejected to the condenser, turbine heat rate, unit capability, and condenser flow a considerable amount of longhand calculations were necessary prior to giving this information to the computer programmers. It would be very desirable if the tables could be developed directly by the computer from the given values.

2. The effect of the cut-off point on the calculated back pressure at low circulating water temperatures has not been incorporated in the study due to the inability of programming a formula for it. Since the amount of cut-off correction is directly related to the amount of air leakage, and since most of the condensers on the system operate with less than 3 to 5 cfm air leakage, it

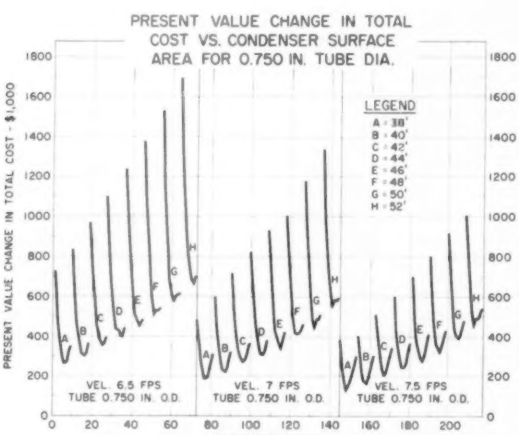


Fig. 5—Similar cost information to that in Fig. 3 comes from a plot of different length condenser loops with different water velocities and in different sized condensers

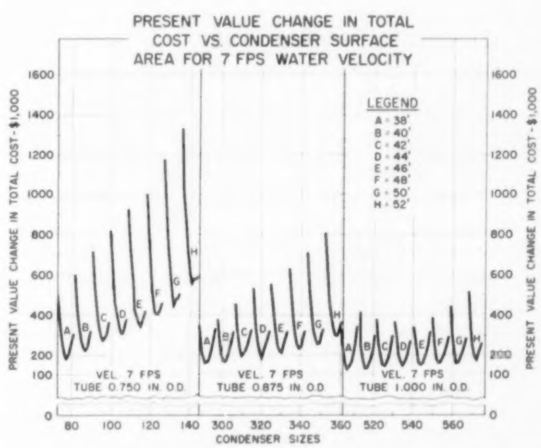


Fig. 6—Still another family of curves yielding valuable computer data is this one showing the effect on costs as surface area, tube length, and tube diameter are varied under a constant water velocity

was reasoned that the effect of the cut-off would be negligible if the spare pumps were also placed in service at such times when a single vacuum pump limited the condenser performance. It would, however, be desirable to be able to insert the cut-off formula in the program.

3. A considerable amount of work is involved in reviewing the print-out and plotting the results to determine the most economical condenser. It would be very helpful if the computer could plot curves of the results or list the condensers by their relative economics.

Conclusions

The results of this study show that the use of a digital computer for a large condenser selection is fully justified.

To obtain optimum range of variables and maximum use of calculation refinements it is necessary to select a computer with ample memory and rapid speed.

The program used by Edison should be expanded to reduce the longhand calculations to a minimum.

A judicious selection of print-out material will aid the designers in their study and understanding of the effects of the condenser variables.

It is important that turbine heat rates, vacuum corrections, condenser flows, and heat rejected be obtained early in the design schedule and that they be accurate. To accomplish this objective, these items should be carefully delineated in the turbine specifications.

Once a condenser problem is well programmed for the computer it is relatively easy to modify it for other condenser studies.

References

(1) Heat Exchange Institute, "Standards for Steam Surface Condensers," 4th Ed., New York: Heat Exchange Institute, Copyright 1955.

(2) Bartlett, R. L., "Steam Turbine Performance and Economics," New York: McGraw-Hill Book Company, Inc., 1958.

Ash Handling Equipment For Industrial Power Plants

The following paper was presented at the American Power Conference in Chicago, March 26–29. **For more details of the Conference see pp. 43.**

A. M. Perrin, National Conveyors Co., confined his paper "Ash Handling Equipment for Industrial Power Plants" to what he held to be the most widely used ash handling system, the steam pneumatic ash conveyor, a typical one of which he illustrated by slide and described its components.

The engineer seeking a good ash conveyor system, Mr. Perrin stated, should acquaint himself with the available equipment by discussing the problem of ash handling with manufacturers of equipment, check their design and inspect a few installations, and avoid in drawing specifications two pitfalls; first—a hybrid specification; and second—an overly-detailed specification. The hybrid specification is one in which the descriptions of different components are based on certain parts made by different manufacturers. This is not desirable because in order to follow the letter of the specification, each manufacturer must then add extra costs to his estimate to incorporate features which are not his standards. As a result, prices are increased.

The overly-detailed specification is one which does not encourage the bidder to include in his quotation recent improvements in design or suggestions that can improve performance and reduce costs. Instead the bidder becomes fearful that any deviation from the specifications might mean disqualification, and as a result the buyer does not always get the best system at the best price.

It is generally good practice to install a system that will remove the ash accumulated in 24-hr operation of the plant in less than 4 hr. If the ash handling equipment is to serve coal burning equipment that will produce large clinkers, the factor which will govern the size of the conveyor will be the size of the clinkers rather than the hourly capacity of the system.

Ash conveyor fittings, which include elbows and laterals, represent approximately 8 per cent of the total material cost. They experience more wear than any of the other components. Ash conveyor pipe can represent 10–15 per cent of the total; the ash storage bin with its discharge gate or ash conditioner unit can represent as high as 30–35 per cent of the total, and the receiving equipment (that part of the ash conveyor system which includes the centrifugal receiver, secondary separator, steam exhauster unit and air washer), represents about 22 per cent of the total material cost of the installation.

Installation costs from the authors' company experience represents about 50 per cent of the material costs for the small system, and 35 per cent of the material costs for the larger system. The ash conveyor manufacturer will install the equipment he furnishes; however, due to scheduling, and the various trades involved, it is usually good practice to have the following work performed by others: Foundations and anchor bolts; trenches and trench cover plates; electric wiring and conduit; steam, water, or compressed air piping; and field painting. The installation of the ash conveyor equipment can also be performed by the customer's contractor or the plant maintenance department. In this case the author recommended that they employ at least the part-time services of the conveyor company's service superintendent to supervise and check the installation and place the entire equipment into regular operation.

Abstracts From the Technical Press—Abroad and Domestic

(Drawn from the Monthly Technical Bulletin, International Combustion, Ltd., London, W. C. 1)

Grinding, Screening and Filtering

Fine Grinding in Mechanically Driven Impact Mills. W. Beushausen. *Chem. Ing. Tech.* 1959, **31** (Sept.), 553–60 (in German).

The features particular to grinding by impact are described and compared with other methods and the fields of application outlined. Various designs of impact mills are illustrated. The relationship between energy consumption and fineness of grinding, effect and influence of air as carrier, the comminution process in the machines, the influence of impact velocity on fineness of grinding and the efficiency of the comminution process by impact are discussed.

Operational Data on a Ring-Ball Pulverizer. F. J. Hiorns and C. P. Sayles. *J. Inst. Fuel* 1959, **32** (Oct.), 464–75.

Tests were carried out on a ring-ball mill with different coals and the results of a statistical analysis of the tests with two coals are presented. The significance of the effects of air rate, classifier and fuel: air ratio settings on fineness and rate of output and power consumption is shown. A simpler analysis is used to relate control board readings to these same factors and recommendations for the operation of such mills are made. Examples of nomograms are given which permit obtaining fineness and rate of output from control board readings; these give the mill operator information that is not normally available.

Electro-Hydraulic Crushing of Coal. B. L. Losev, A. N. Mel'Nikova, F. Ya. Saprykin and L. A. Yutkin. *Vestnik Akad. Nauk SSSR.* 1959 (6) 62–5. *LLU Transl.* 1959 (Oct.), 34–8.

In this process electrical energy is directly converted into mechanical energy by passing a continuous succession of high voltage transient electrical discharges with very steep front and short current-impulse through water in a tank in which the coal is submerged. Two hydraulic shocks are generated, a primary effect and a cavitational; in the first the spark channel enlarges and forms an expanding cavity which then collapses at ultrasonic speed and generates a second hydraulic shock. Results obtained with Moscow bituminous and brown coal are given. The ash and sulfur contents of the coal were reduced but its chemical composition was not changed, nor was it oxidized to a significant extent.

Capacity, Separating Effect and Dimensions of Solid-Bowl Centrifuges. H. F. Trawinski. *Chem. Ing. Tech.* 1959, **31** (Oct.), 661–6 (in German).

Phase segregation in the centrifuge is governed by Stokes' law by which the suspension is characterized by the settling velocity in a gravity field. The controlling parameters are the bowl surface and the centrifugal acceleration. In these the length, diameter and rate of rotation are contained. Diameter and rate of rotation can be eliminated by introducing the tensile strength to spec. gravity ratio of the material. It is shown that the separation effect of the rotor can be increased only by increasing its length but not its diameter.

Radiometric Absorption Analysis. I. Possibilities of Chemical Ultimate Analysis with a Methane Flow Counter as Ray Detector. L. Wiesner. *BrennstChemie* 1959, **40** (Sept.) 273–8 (in German).

The advantages of the methane counter over the Geiger-Müller counter are discussed and the instrument and its application are described. The quantitative determination of hydrogen, oxygen, phosphorous, sulfur, lead, chlorine and iodine is presented; simultaneous determination of several elements is possible. The question of costs is discussed.

Analysis and Testing

The Differential Thermal Analysis as a Method of Evaluation of Coals. C. Kröger, H. Hovestadt and E. Bade. *BrennstChemie* 1959, **40** (Sept.), 286–9 (in German).

The influence of inert, oxidizing and salt forming oxides and of iron and copper on the behavior of macer-

als during differential thermal analysis is described. Characteristic effects are obtained with cuprous oxide which make possible a characterization of the coal substance. The mineral content of coals, especially of enriched fractions, can be determined by differential thermal analysis faster and almost as far as with X-ray analysis.

Problems of Air Pollution in Germany. H. Lent. *J. Inst. Fuel* 1959, **32** (Oct.), 485–501.

The ways in which the air pollution problem is being tackled in Germany are outlined. Pollution caused by large and small industrial power plants and by domestic heating is discussed. Sulfur dioxide is regarded as the most common pollutant and recent evidence of damage to flora and fauna caused by SO_2 cited. The task of the Clean Air Commission set up in Germany is defined.

Instruments and Controls

Engineering Design by Digital Computers. B. J. Chalmers. *Research* 1959, **12** (Oct./Nov.) 390–4.

The application of digital computers to design problems in general and their ability to optimize a design owing to their discrimination facility are presented. It is shown that computers can be used for both design details and complete designs and that results are obtained in a fraction of the time needed by human effort.

Electronic Boiler Control. Anon. *Elect. Rev.* 1959, **165** (Nov. 13) 635–42.

At Little Barford "B" station an electronic control system has been installed in which the controlling parameters are the steam pressure and air/steam flow at the superheater outlet. Transmitters, three-term process controllers, single analogue computers and electro-hydraulic positioners control combustion, steam temperature and mill feed. The system is described in detail.

A New Sensitive Temperature Detector for Use in High Pressure Fluid Piping. D. Robertson. *A.S.M.E. Preprint* No. 59-A-201 1959, (Dec.) 7 pp.

A new composite thermocouple in a swaged plug has been developed for measuring temperature changes in the steam flowing through a heavy-wall pipe. Details of design and response time are given.

A Thermocouple for Reactor Control. J. L. Ayre. *Nucl. Pwr.* 1959, **4** (Dec.) 117–8.

The design of the thermocouple for measuring the outlet temperatures of the gas and the tests made to find the

best way of its installation in the channel are described.

Radioactive Level Gauges. J. F. Cameron and E. W. Pulsford. *Brit. Chem. Engng.* 1959, **4** (Nov.) 605–7.

Instruments employing radioactive sources for detecting the liquid level inside tubes, tanks and drums are described.

Nuclear Energy

Neutron Research. Anon. *Nucl. Energy Engr.* 1959, **13** (Oct.), 491–4.

Research carried out at the U. S. Bureau of Standards and the establishment of a national neutron standard are described. Details are given of improvements in the accuracy of neutron measurements, neutron cross-section measurements, selection of sources, "button" detectors, γ-ray dosimeters and the checking of theory by experiments.

Approximate Calculation of the Surface Loss of a Reactor. M. Ledinegg. *Atomkernenergie* 1959, **4** (Sept.), 351–6 (in German).

Good agreement has been found in calculating neutron losses by the simple diffusion law and the two-group theory, the former being much simpler and less time consuming.

Perturbation Methods for Reactor Diffusion Equations. L. H. Holway. *Nucl. Sci. Engng.* 1959, **6** (Sept.) 191–201.

A new method for solving the diffusion equations has been developed and is illustrated by examples.

Heat Stresses in Steel Walls of Reactor Pressure Vessels under Neutron Bombardment and One-Sided Cooling. G. Grass. *Atomkernenergie* 1959, **4** (Sept.), 364–8 (in German).

Equations are derived for the temperature gradient and thermal stresses in steel walls under neutron bombardment as a function of number and energy of the neutrons, steel properties and vessel dimensions. The results are given in tables and graphs.

Structural Design in High-Temperature Reactors. N. H. Triner. *Nucleonics* 1959, **17** (Oct.), 78–81.

For the design of high-temperature reactors more emphasis must be put on experimental investigations into the load stress, displacement stress, thermal stress and creep of the materials to be used to be sure that the calculations are based on valid data.

Atomic Energy and Public Health. Anon. *Atom* 1959, (Oct.) 9–11.

In his address to the Cambridge conference of the Food and Agriculture Organization of the United

Nations Sir John Cockroft spoke of the danger of leakage from air-cooled reactors (escape of iodine-131, strontium-90, strontium-89 and caesium-137), gas-cooled reactors and fast reactors and efforts to reduce hazards. He stressed the many variables involved in the spread of these poisons and their uptake by animals and humans, the necessity for reliable measurements, and international cooperation in this field and finally discussed discharge of radioactive effluent into rivers and storage methods.

Atomic Review. Thermonuclear Revival. Anon. *Engineering* 1959, **188** (Oct. 30) 406–7.

Brief abstracts are presented of the papers to the Uppsala conference on ionization phenomena and the London conference on very high temperature physics, both topics being related to research into thermonuclear fusion.

The Industrial Future for Large Radiation Sources. S. Jefferson. *Nucl. Pwr.* 1959, **4** (Nov.) 104–5.

Useful applications of large sources of radiation were discussed at a conference in Warsaw in September 1959. A survey of the proceedings is presented.

Oxidation of Graphite in Reactors. M. Tomlinson. *Nucl. Pwr.* 1959, **4** (Nov.) 117–8.

The reasons for the large discrepancies found in recent publications are discussed. It is shown that reasonable agreement is possible if allowance is made for the differences in the parameters used.

Thermal Instability Due to Graphite Oxidation. P. J. Robinson and J. C. Taylor. *Nucl. Engng.* 1959, **4** (Nov.) 400–3.

It is shown that once the heat generated by oxidation of the graphite exceeds that lost to the surroundings the rise of temperature is rapid and leads to fires difficult to extinguish.

Temperature Distribution in Double-Flow Reactor Cooling Channels. II. H. D. Baehr and W. Strewe. *Atomkernenergie* 1959, **4** (Oct.) 398–402 (in German).

The second part compares parallel and counterflow with regard to heat transfer and pressure drop and shows that counterflow heat transfer is far superior to parallel flow but that this advantage may be nullified by the greatly increased pressure drop. In reactors cooled by superheated steam or liquids the higher pressure drop is of lesser importance than in gas-cooled reactors. A calculated example shows these effects.

A New Method for the Calculation of Thermal Utilization in Heterogeneous Reactors. A. Kirchenmayer. *Atomkernenergie* 1959, **4** (Oct.) 395–7 (in German).

A method for simplifying the calculation is described.

The Thermal Utilization of the Heterogeneous Reactor in the Light of the Two-group Theory. A. Kirchenmayer. *Atomkernenergie* 1959, **4** (Oct.) 397–8 (in German).

The application of the new method of calculation described in the previous article is presented.

Calculation of Fast Neutron Multiplication in Thick Fuel Elements. K. O. Thielheim. *Atomkernenergie* 1959, **4** (Nov.) 429–37 (in German).

Fast neutron multiplication has been calculated by means of a digital computer taking into account change of neutron collision probability in the course of a chain reaction initiated by a fast primary neutron and the reduction of kinetic energy of fast neutrons by inelastic scatter on fuel atom nuclei and their influence on multiplication. In thin fuel elements both these factors need not be taken into account. The results are tabulated.

The Power of a Heavy Water Reactor During an Emergency Shutdown. I. M. El-Ibiary. *Nucl. Energy Engr.* 1959, **13** (Nov.) 545–9.

Experimental results and those obtained by a simulator are compared and from the best fit of various theoretical solutions the value of the shut down reactivity determined.

The Breeder Goes Critical. Anon. *Nucl. Pwr.* 1959, **4** (Dec.) 93.

The Dounreay fast breeder reactor (15 MWe) went critical on November 14, 1959. The main reactor data are given.

Commissioning and Start-Up. J. L. Phillips. *Nucl. Pwr.* 1959, **4** (Dec.) 94–100.

The steps leading up to the commissioning of the Dounreay reactor are detailed: 1. Construction tests; 2. Staff training; 3. Commissioning: (*a*) dry testing; (*b*) metal charging; (*c*) wet testing; (*d*) final approach.

Instrumentation and Control. K. R. Sandiford. *Nucl. Pwr.* 1959, **4** (Dec.) 100a–102a.

The instruments for neutron flux, coolant flow and temperature measurement and the control system based on the insertion of 12 mobile fuel element rods from below and three boron rods from above are described.

Behavior of Ultrasonics in the Presence of Various Types of Defects Which May Be Found in Welds. P. Bastien and M. Evrard. *Soudage* 1959, **13** (Sept./Oct.) 347–57 (in French).

Tests were carried out by various laboratories on identical pieces exhibiting three types of defects, lack of penetration, blow holes and slag inclusions. The results have led to the determination of the behavior of ultrasonics with respect to the typical defects considered and the definition of the nature of these defects from the form of the recorded oscillograms.

A Test to Anticipate the Performance of Steam Piping at 5000 Psi and 1200 F. G. C. Wiedersum. *A.S.M.E. Preprint* 59-A-230 1959, (Dec.) 8 pp.

A test rig has been set up containing a length of welded pipe of the material used in the plant in which operating conditions are simulated as nearly as possible and periodic checks can be made of the changes in the weld and material due to thermal and internal pressure stresses. The rig and the type of tests to be made are described.

Fuel Research. Annual Report for Year Ending 30th June, 1959. Commonwealth Scientific and Industrial Research Organisation. Coal Research Section. P.O. Box 3, Chatswood N.S.W. 1959, 18 pp.

The research undertaken by the Coal Research Section is outlined under the headings: 1. General; 2. Coal Utilization: (*a*) carbonization; (*b*) chemicals from coal; (*c*) combustion and gasification; 3. Examination of coal seams.

Fuels: Sources, Properties and Preparation

Factors to Consider in the Use and Application of Industrial Coals. U. B. Yeager. *ASME Preprint* No. 59-Fu-2, 1959 (Oct.), 9 pp.

Coal properties and their effects are discussed in relation to the firing equipment intended to be used. It is stressed that in almost all cases a washed coal will be more economical than a run-of-mine coal.

Effect of Coal Characteristics on Combustion Performance. T. S. Spicer. *Coal Utiliz.* 1959, **13** (Nov.), 19–23.

The coals most suitable for single-retort, multiple-retort, spreader stoker, traveling grate, pulverized fuel and cyclone firing are discussed and the properties most important for each type of firing tabulated.

High Pressure Water Loop in Dido.

I. Everson. *Nucl. Pwr.* 1959, **4** (Dec.) 103–8.

An account is given of the installation of the loop in the reactor, measurements and safety arrangements, leakage from the loop and welding precautions, in-pile test section, valves and commissioning tests. These latter consisted in measurements of leakage rate, in testing the emergency blowdown system and instrumentation, and in establishing a start-up procedure.

Grinding, Screening and Filtering

Some Experience of Coal Grinding Plant Tests on a Ring-Ball Mill and a Tube Mill. A. Fitton and R. Jackson. *J. Inst. Fuel* 1959, **32** (Nov. 520–9.

The results obtained with these two types of mills tested (ring-ball mill at Little Barford and tube mill at Brimsdown) and published earlier have now been compared and some conclusions drawn of power consumption, drying capacity, fineness of grinding and output. The ring-ball mill appears to be more suitable for varying loads, the tube mill for steady conditions at high loads; the drying capacity and fineness of grinding of the former is higher. Power consumption shows little difference.

The Effect of the Speed of Rotation and the Design of the Lining on the Performance of a Drum-type Ball Mill. B. L. Koutman. *Teploenergetika* 1959, (Nov.) 24–8 (in Russian).

The results of experimental investigations are presented and recommendations made on the speed of rotation and the design of the mill lining.

Analysis and Testing, Research

Analytical Determination of Moisture Content of Brown Coal and Low Temperature Brown Coal Coke by Drying with High Frequency Heating. J. Jandasek. *BrennstChemie* 1959, **40** (Oct.) 309–14 (in German).

Tests are described which had the following results: 1. Moisture determination of brown coal and moist coke by difference using high frequency heating (17Mc/s, 3–6 kV) was sufficiently accurate and agreed well with the xylol distillation method; 2. The time required was about 15 min; 3. The method is specially useful for series determinations; 4. Avoidance of oxidation and of chemicals are added advantages. The procedure used is described in detail.

Penetrants for Non-Destructive testing. R. Schnurmann. *Engineering* 1959, **188** (Nov. 20) 515.

A brief review of the techniques and

Relationship of Coal Ash Viscosity to Chemical Composition. W. L. Sage and J. B. McIlroy. COMBUSTION 1959, **31** (Nov.), 41–8.

Two methods are described for assessing the suitability of a coal for use in a slagging furnace. One is the calculation of the USBM equivalent silica percentage in which the ferric percentage is also calculated by a given equation, and which gives an indication of the temperature of the critical viscosity. The other is the basic-to-acid-ratio method which is carried out at or corrected to 20 ferric percentage. This latter method is specially useful for calculating the amount of flux required to obtain a desired viscosity.

The Suitability of Foreign and Home-produced Fuels for the Operation of Slagging Furnaces. Pt. I. Bituminous Coals. T. Geissler. *Energie* 1959, **11** (Nov.), 526–34 (in German).

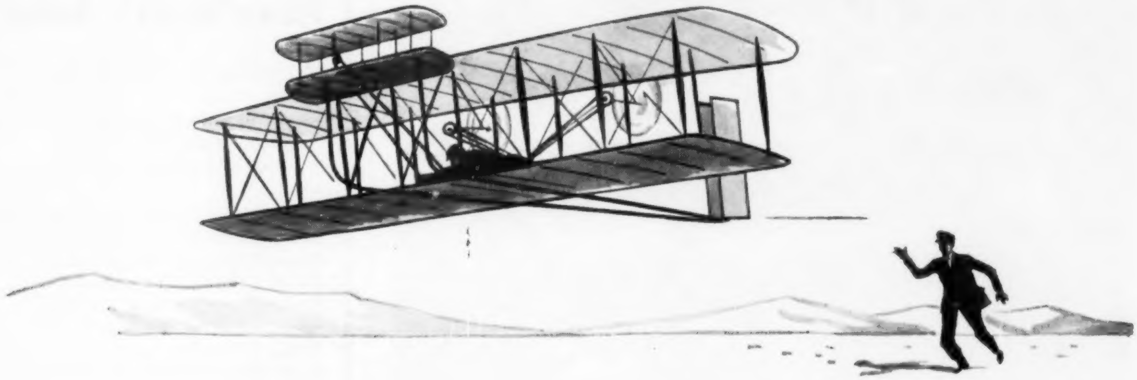
Based on large-scale investigations an equation has been established for the calculation of a coal coefficient K_s from which the suitability or otherwise of a coal for slagging furnace operation can be deduced. The equation is: $K_s = K_1.K_2.K_3.K_4.K_6 + K_5$, where K_1 is dependent on ash content, K_2 on moisture, K_3 on volatiles, K_4 on ash fusion point, K_5 on air preheat temperature with V.M. as parameter, and K_6 on boiler output. The values of these coefficients can be taken from graphs. Values of K_s: from 0.7–1.2 (Class I) very suitable, 0.5–0.7 (Class II) suitable, 0.3–0.5 (Class III) just suitable, 0.2–0.3 (Class IV) difficult operation, 0.1–0.2 (Class V) very difficult operation, 0–0.1 (Class VI) unsuitable. The influence of slagging furnace design is to be discussed in a following article.

The Removal of Chlorides from Coal by Leaching. I. Laboratory Experiments. G. N. Daybell and E. W. F. Gillham. *J. Inst. Fuel* 1959, **32** (Dec.), 589–96.

The laboratory tests have shown that the chlorine content of slacks can be materially reduced in a very short time by washing with water. It is suggested that coal of high chlorine content could be transformed into a harmless fuel by exposing it in a thin layer to weathering (rain).

Further Studies on the Mechanism of the Oxidation of Coal. B. K. Mazumdar, S. K. Chakrabartty, M. Saha, K. S. Anand and A. Lahiri. *Fuel* 1959, **38** (Oct.), 469–82.

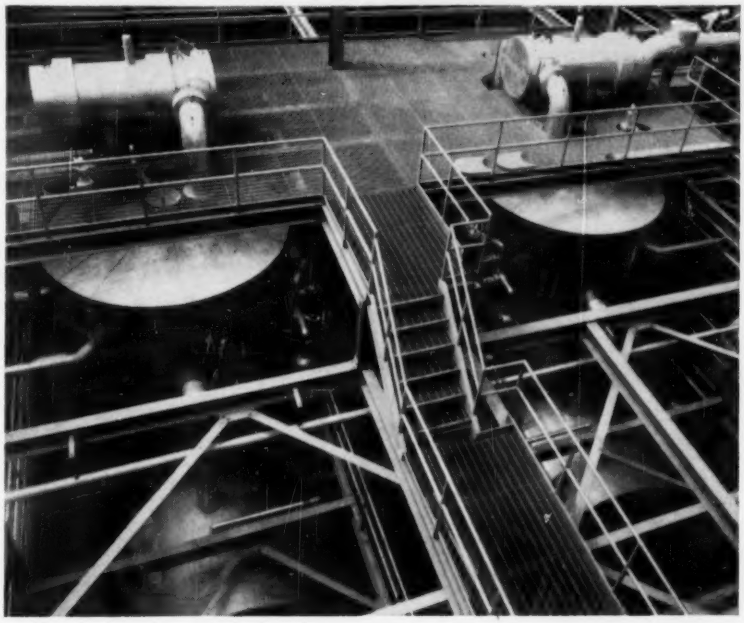
The studies indicate two stages in coal oxidation: below 200 C the coal structure is reduced to its aromatic skeleton, above 200 C rupture of the rings starts.



Advertisers' Index

(*Continued on page 63*)

FLUIDICS* AT WORK



How they're saving feedwater dollars at L.A.'s new Scattergood Steam Plant

Each of these two 12'-diameter, 1,036,250-lb/hr-capacity Permutit spray-type deaerators, nests atop its own 160,000-lb. storage tank in the new Scattergood Steam Plant at Los Angeles. The only open direct-contact-type heater with five closed heaters operating on the regenerative cycle, each deaerator operates at 20 psia to 52.8 psia. Design pressure is 100 psig, ASME code.

These deaerators process boiler feedwater for normal throttle steam conditions of 1800 psig at 1000° F with 1000° F reheat.

The City of Los Angeles can't afford to have corrosive oxygen or CO_2 in that water.

And it doesn't, because Permutit deaerators are *guaranteed* to remove all free CO_2 and to reduce oxygen to less than 0.005 m. per liter.

Save money

These deaerators save operating dollars because they have several advantages over tray or other types of deaerators. Maintenance is minimized, due to simplicity of design. Uniform, wasteless distribution of steam through water in scrubber, at *all* rates of flow from zero to full capacity. No water hammer, no rumbling—in short, quiet operation.

Besides these heaters, two more Permutit deaerators are used at Scattergood to prepare feedwater for evaporators.

Ultimately, Scattergood will have four generating units producing a total of 1,200,000 KW, almost equalling Hoover Dam. There's no room for boiler troubles in this kind of operation.

30-page bulletin

You can get the same feedwater purity and dollar savings, with Permutit deaerators (3000 to 3,000,000 lb/hr) and other water treatment facilities available through our Fluidics program. To start, send for our fact-filled 30-page bulletin, "Spray Type Deaerating Heater."

For more information, write PERMUTIT Division, Dept. CO-50, 50 West 44th Street, New York 36, N. Y., or call your local Permutit representative. In Canada: The Permutit Company of Canada, Ltd., 207 Queens Quay West, Toronto, Ontario.

*FLUIDICS is a complete, imaginative Pfaudler Permutit program which economically provides practical answers to problems involving fluids and gases.

PFAUDLER PERMUTIT INC.

Specialists in FLUIDICS . . . the science of fluid processes